Agriculture Canada

Handling Agricultural Materials

Air and pneumatic conveyors

Canada

Handling Agricultural Materials

Air and pneumatic conveyors

Research Branch
Agriculture Canada

Publication 1832/E
1989

Cat. No. A15-1832/1989E
ISBN 0-660-13086-6
Printed 1989

Available in Canada through authorized bookstore agents and other bookstores or by mail from

Canadian Government Publishing Centre
Supply and Services Canada
Ottawa, Ont. Canada K1A 0S9

Price is subject to change without notice

Previously published under the title
Canadian Agricultural Materials Handling Manual.
2.2. Air and Pneumatic Conveyors.
Agric. Can. Publ. 5002

Canadian Cataloguing in Publication Data

Handling agricultural materials : air and pneumatic conveyors

(Publication ; 1832)

Previously published as: Canadian agricultural materials handling manual. 2.2. Air and pneumatic conveyors.
Issued also in French under title: Manutention de produits agricoles : appareils de déplacement d'air et transporteurs pneumatiques.
Prepared by UMA Engineering Ltd. cf. p. 5.
Bibliography: p.
Cat. no. A15-1832/1989E
ISBN 0-660-13086-6

1. Conveying machinery. 2. Agricultural machinery. 3. Pneumatic machinery. I. Rudnitski, Sharon. II. Birenbaum, Rhonda. III. Canada. Agriculture Canada. Research Branch. IV. UMA Engineering Ltd. V. Series: Publication (Canada. Agriculture Canada). English ; 1832.

TJ1350H3 1989 621.8'67 C89-099201-0

Staff editor
Sharon Rudnitski
Research Program Service

Contract editor
Rhonda Birenbaum

Scientific adviser for contract research
L. Heslop
Engineering and Statistical Research Centre

CONTENTS

Figures

FOREWORD

Handling Agricultural Materials is produced in several parts as a guide to designers of materials-handling systems for farms and associated industries. Sections deal with selection and design of specific types of equipment for materials handling and processing. Items may be required to function independently or as components of a system. The design of a complete system may require information from several sections of the manual.

Air and Pneumatic Conveyors was prepared by UMA Engineering Ltd., Winnipeg, Man., for the Canada Committee on Agricultural Engineering Services of the Canadian Agricultural Services Coordinating Committee.

1 # GENERAL

1.1 ## Air conveyors

Fans, blowers, and compressors are the most widely used air-moving devices in Canadian agricultural facilities. The effective use of these types of conveyors, however, requires an understanding of the properties of air plus knowledge of the various components and types of air-handling systems.

1.2 ## Pneumatic conveyors

Pneumatic conveyors, a flexible means of handling grain, are useful where space is limited and where the conveying route changes direction many times. More and more farm and commercial operations rely on pneumatic conveyors to handle grain, feed, and some fertilizer products.

2 # PROPERTIES OF AIR AND CONDUIT

2.1 ## Properties and behavior of air

Atmospheric air consists of a mixture of gases, mainly nitrogen and oxygen, plus water vapor. Standard air, used in rating fans and blowers, has a density of 1.2 kg/m^3 at a temperature of 20°C and an atmospheric pressure of 100 kPa.

Since air obeys the perfect gas laws, its density varies with changes in pressure and temperature. However, changes in density have little effect on air transport systems, except as this characteristic relates to the fan power requirements.

In designing systems to handle agricultural materials, base the power requirements on ambient winter temperatures. An air-moving system handling air at –40°C, for example, requires 26% more power than the same machine operating at standard conditions. Fan manufacturers provide correction factors to allow for variations in both atmospheric pressure and ambient temperatures (Table 1).

Although air is a compressible gas, in low-pressure systems (around 10% of atmospheric pressure) treat it as incompressible. In all other situations, however, do not overlook its compressibility.

Atmospheric moisture does not have a significant effect on fan performance. High moisture content may slightly reduce power requirements, but this effect can generally be ignored.

2.2 ## Properties of the air-handling conduit

Be careful in selecting the ducting for air-handling systems, whether for dust collection or material transport. Choose only round ducts because their ratio of flow area to friction surface is large. Use square-to-round transitions at dust pickup points and at the discharge to material collectors.

Construct ductwork with longitudinal lock-welded seams or with spiral construction.

Form elbows with long radii. Mitred elbows are not recommended. The choice of construction materials depends on the materials handled and the corrosiveness of the plant atmosphere.

Rely on the standards of the Sheet Metal and Air Conditioning Contractors' National Association (SMACNA) or the American Society of Heating, Refrigeration, and Air-Conditioning Engineers (ASHRAE) for information on duct wall thicknesses. Increase wall thickness, however, for ducts handling abrasive materials or operating continuously.

Do not use nonmetallic materials in systems conveying agricultural products. Such nonconductive materials can build up excessive static electrical charge, possibly causing personal injury or dust explosions. The Dominion Fire Commissioner standards for grain elevators state that all spouting and ducting must be electrically conductive and grounded. One notable exception to this rule is the plastic pipe commonly used for filling tower silos. The high moisture content of the silage prevents the buildup of static charge in these situations.

3 # FANS AND FAN SYSTEMS

3.1 ## Selecting a fan

Use this information to select a fan:

- air flow rate
- fan static pressure
- type of material expected to pass through the fan:
  a) fibrous materials – choose a fan able to handle heavy dust loads
  b) clean air – choose an ordinary service fan
  c) corrosive material

Table 1 Density correction factor

Temp., °C	Altitude, metres above sea level											
	–250	0	250	500	750	1000	1250	1500	1750	2000	2500	3000
–40	1.30	1.26	1.23	1.18	1.14	1.10	1.06	1.00	0.98	0.94	0.90	0.87
–18	1.18	1.15	1.12	1.08	1.04	1.00	0.96	0.92	0.90	0.86	0.82	0.79
0	1.11	1.08	1.05	1.02	0.99	0.96	0.93	0.91	0.88	0.86	0.81	0.76
21	1.03	1.00	0.97	0.95	0.92	0.89	0.87	0.84	0.82	0.79	0.75	0.71
50	0.94	0.91	0.89	0.86	0.84	0.81	0.79	0.77	0.75	0.72	0.68	0.64
75	0.87	0.85	0.82	0.80	0.78	0.75	0.73	0.71	0.69	0.67	0.63	0.60
100	0.81	0.79	0.77	0.75	0.72	0.70	0.68	0.66	0.65	0.63	0.59	0.56
150	0.72	0.70	0.68	0.66	0.64	0.62	0.60	0.59	0.57	0.55	0.52	0.49
200	0.64	0.62	0.61	0.59	0.57	0.56	0.54	0.52	0.51	0.49	0.47	0.44
250	0.58	0.56	0.55	0.53	0.52	0.50	0.49	0.47	0.46	0.45	0.42	0.40
300	0.53	0.51	0.50	0.49	0.47	0.46	0.45	0.43	0.42	0.41	0.38	0.36
350	0.49	0.47	0.46	0.45	0.43	0.42	0.41	0.40	0.39	0.38	0.35	0.33
400	0.45	0.44	0.43	0.41	0.40	0.39	0.38	0.37	0.36	0.35	0.33	0.31
450	0.42	0.41	0.40	0.38	0.37	0.36	0.35	0.34	0.33	0.32	0.30	0.29
500	0.39	0.38	0.37	0.36	0.35	0.34	0.33	0.32	0.31	0.30	0.28	0.27

kg/m^3 = density factor × 1.2
mass of dry air at 21°C and sea level = 1.2 kg/m^3
Source: *Industrial ventilation, a manual of recommended practice.*

   d) explosive or inflammable materials—choose a fan made of nonsparking aluminum

- type of motor drive (direct drive or belt driven)
- space limitations
- noise level
- operating temperature and altitude
- operating speed

Manufacturers' literature can help identify the air flow rate and pressure capabilities, as well as pressure and speed limits of the available fans. For greatest efficiency, select a fan that operates in the middle third of the air flow rates available (Fig. 1).

Fan performance curves, which illustrate air flow rates for various pressures, are sometimes available from fan manufacturers. These graphs are useful in understanding how fan characteristics influence system designs.

Fan performance varies with rotational speed. Likewise, a duct system operating at a specific air flow rate performs with a related resistance.

Plotting the graphs of fan and system performance reveals the point of optimal operation at the intersection of these performance curves. At a given air flow rate, the pressure delivered by the fan must equal the pressure developed in the system (Fig. 2).

3.2 Fan performance laws

Many laws relate to fan performance and involve fan size, speed, pressure, and volume. Three performance laws are particularly important to system designers.

- Air flow varies with fan speed.
- Static pressure and total pressure vary with the square of the fan speed.
- Power varies with the cube of the fan speed.

Fig. 3 shows the effect of a 10% increase in fan speed on air flow, pressure, and power. Fig. 4 illustrates the effect of a 20% increase in air flow on system pressure.

For exhaust or pneumatic conveying systems, select a fan that yields little variation in air flow rate with changing system pressure.

3.3 Fan operation: parallel and series

Fans operating in parallel deliver an air flow rate equal to the sum of the individual fan flow rates at points of equal pressure (Fig. 5). Fans operating in series develop a total pressure equal to the sum of the individual fan pressures at points of equal flow rate (Fig. 6). In establishing system specifications in either case, designers must account for losses in individual fan connections.

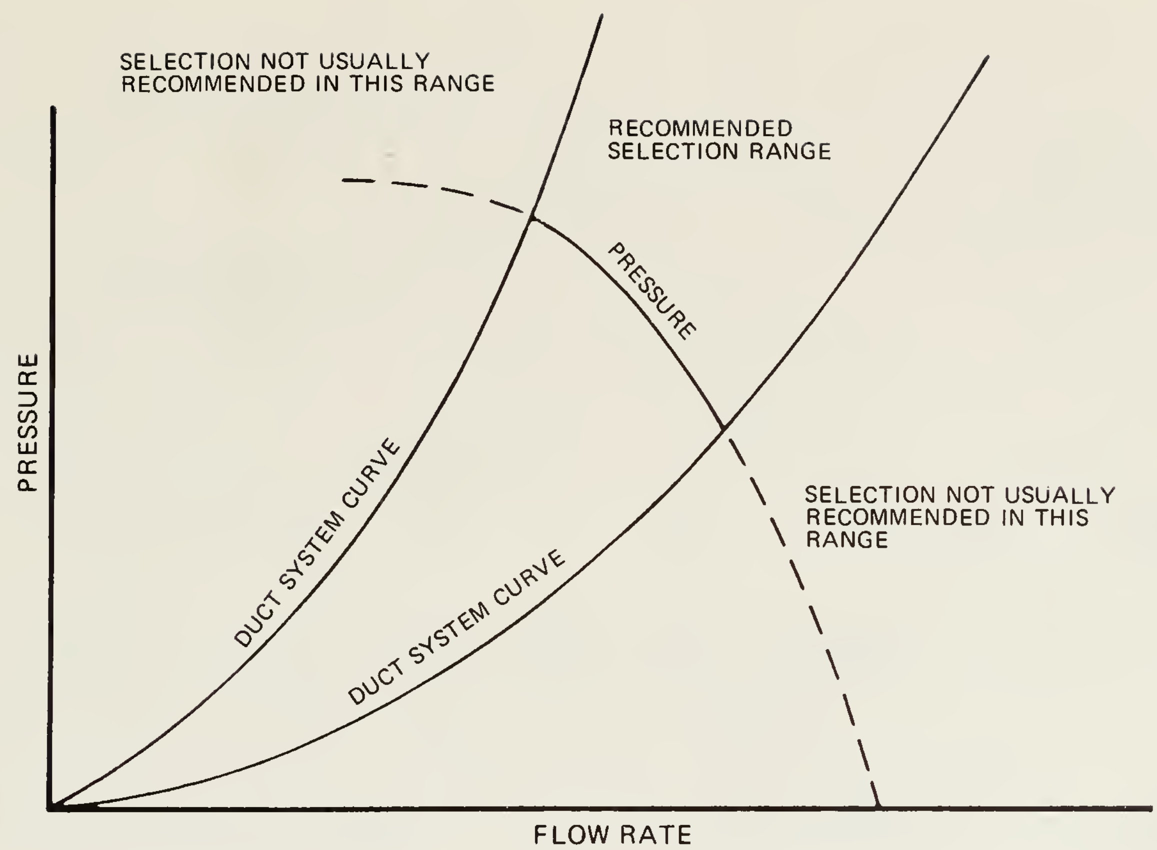


Fig. 1. Recommended performance of a typical centrifugal fan.
Source*: *Industrial ventilation, a manual of recommended practice.

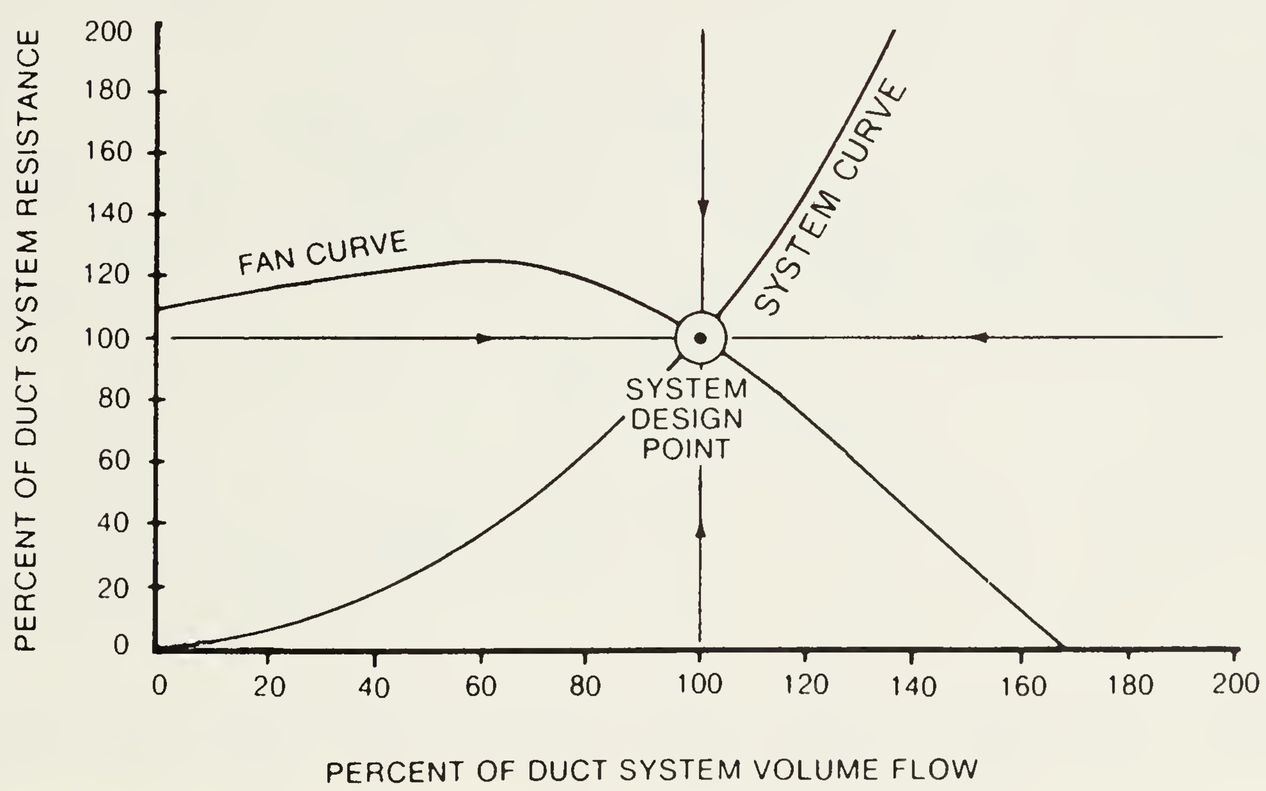


Fig. 2. Interaction of system curves and fan curve.
Source*: *Fan application manual: Fans and fan systems.

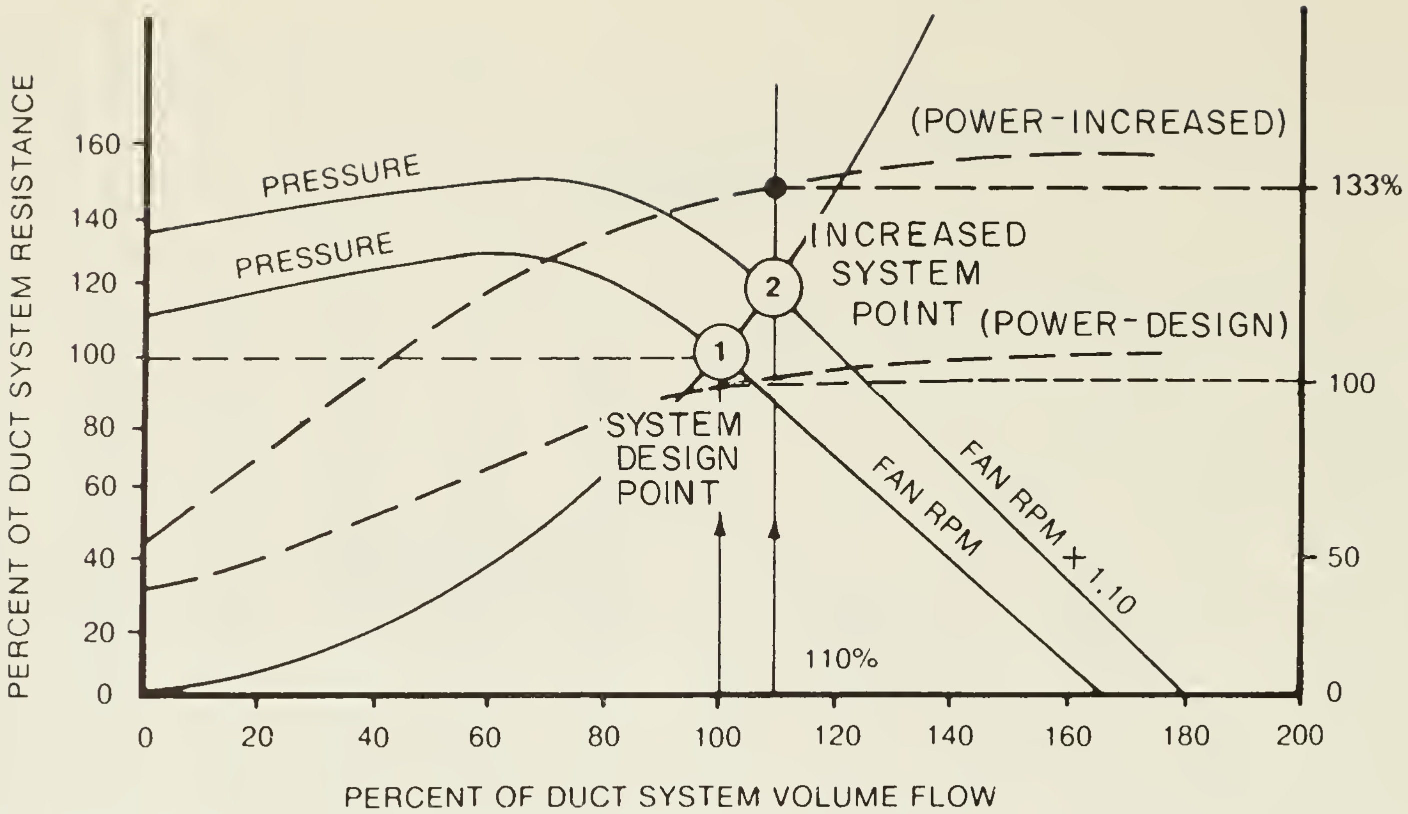


Fig. 3. Effect of 10% increase in fan speed.
Source: Fan application manual: Fans and fan systems.

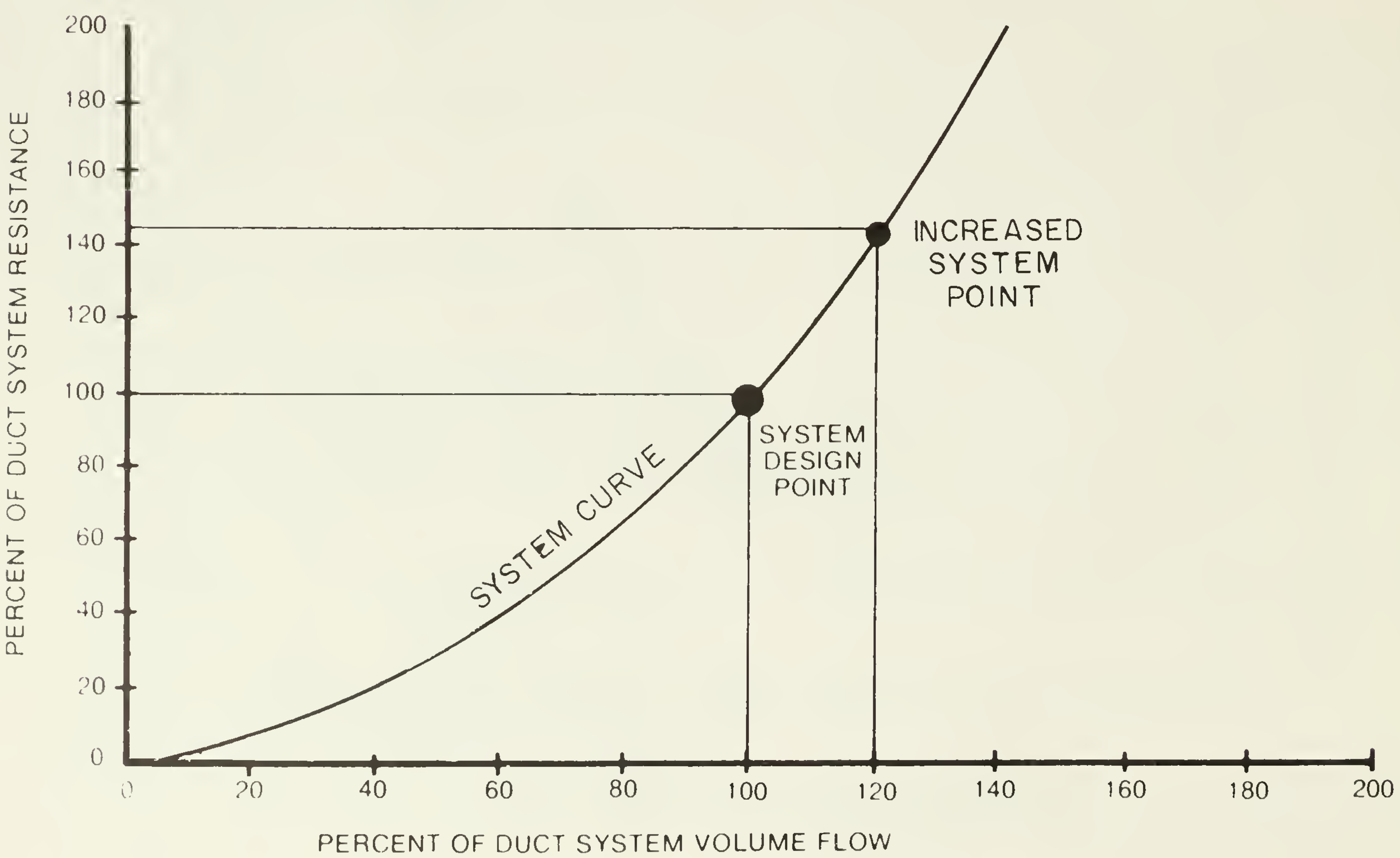
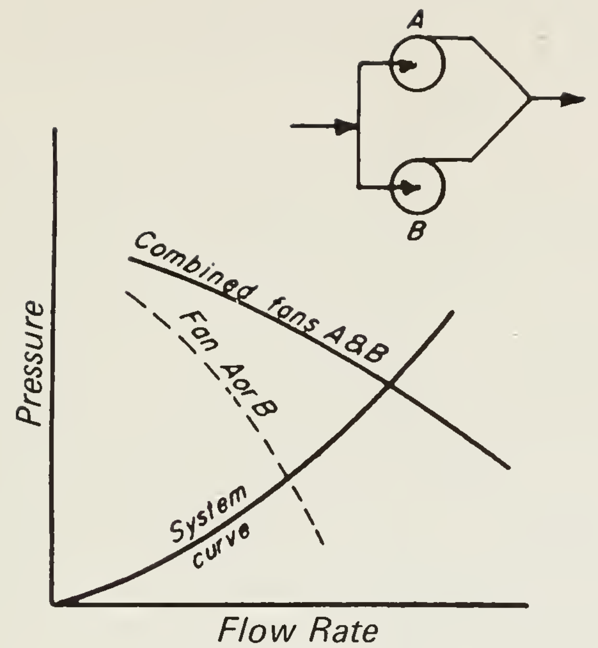
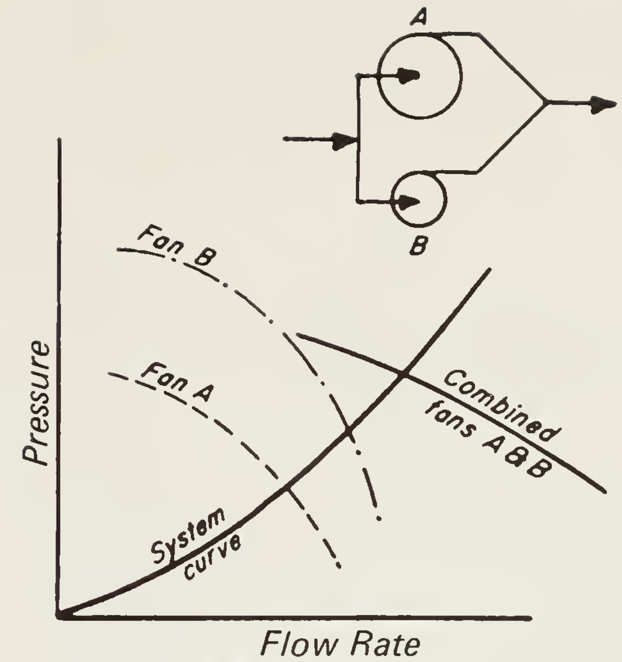


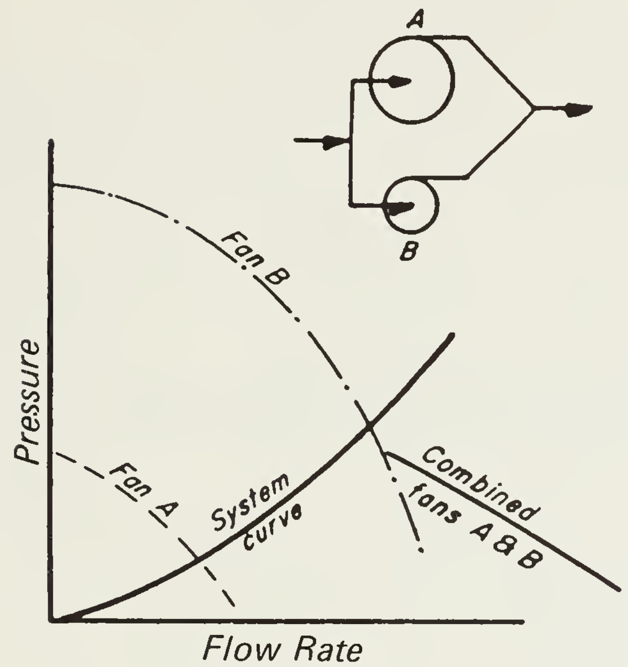
Fig. 4. Effect of 20% increase in flow rate on system pressure.
Source: Fan application manual: Fans and fan systems.



Two identical fans
Recommended for best efficiency



Two different fans
Satisfactory



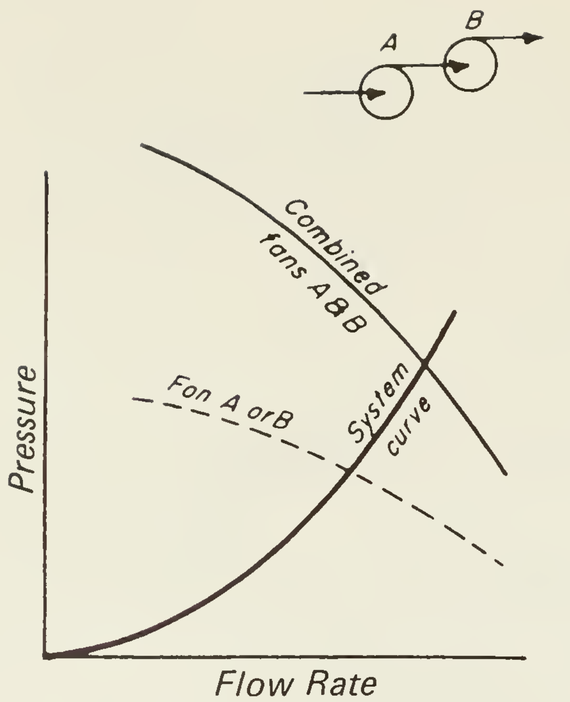
Two different fans
Unsatisfactory

When system curve does not cross combined fan curve, or crosses projected combined curve before fan B, fan B will handle more air than fans A and B in parallel.

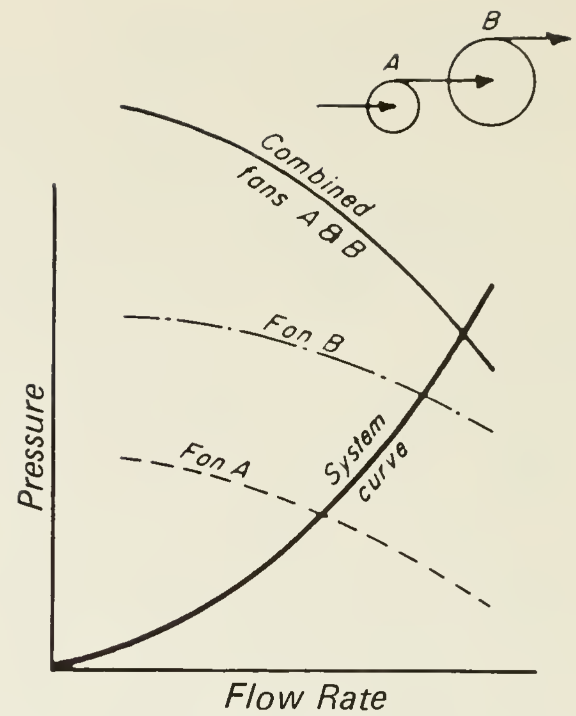
Note:

1. System curve must intersect combined fan curve or higher pressure fan may handle more air alone.

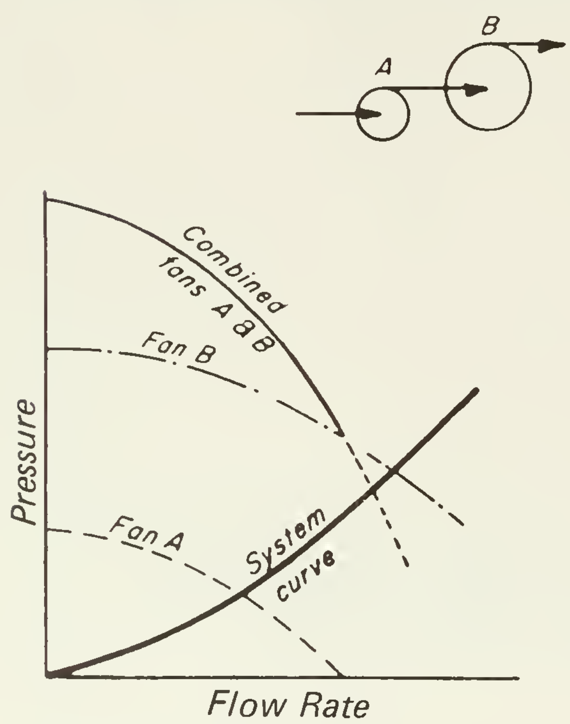
Fig. 5. Fans in parallel operation.
Source: Industrial ventilation, a manual of recommended practice.



Two identical fans
Recommended for best efficiency



Two different fans
Satisfactory



Two different fans
Unsatisfactory

When system curve does not intersect combined fan curve, or crosses projected combined curve before fan B curve, fan B will move more air than fan A and B in series.

Notes:

1. *System curve must intersect combined fan curve or large volume fan may handle more air alone.*

2. *Air volume through each fan will be the same, since air is considered incompressible.*

Fig. 6. Fans in series operation.
Source: *Industrial ventilation, a manual of recommended practice.*

Identical fans yield the best results. When using dissimilar fans, exercise care in their selection to ensure each fan delivers the expected output under the predicted system operating conditions.

3.4 System pressures

Two distinct pressures act to compress or expand fluid flowing through a duct or pipe: static pressure and velocity pressure. Static pressure, which can be negative or positive, represents the differential between the pressure exerted perpendicular to the direction of flow and atmospheric pressure. Velocity pressure, measured in the direction of the flow, is the kinematic pressure necessary to maintain the fluid at a constant velocity.

For a given flow of air or gas in a conduit, total pressure can be expressed algebraically.

$$P_t = P_s + P_v$$

where P_t = total pressure (kPa)

P_s = static pressure (kPa)

P_v = velocity pressure (kPa)

Total pressure, velocity pressure, and static pressure can be measured with such instruments as pitot tubes or velometers. Measure total pressure in the direction of flow.

For more information on determining pressures, consult such publications as *Industrial Ventilation*, published by the American Conference of Governmental Industrial Hygienists, and the *ASHRAE Handbook*, published by the American Society of Heating, Refrigeration, and Air Conditioning Engineers.

3.5 *Static pressure* Static pressure is the pressure required to overcome the resistance to flow. Hence, some publications refer to static pressure as frictional or resistance pressure.

3.6 *Velocity pressure* In pneumatic conveying, velocity pressure is presented in two ways. Firstly, it is the pressure required to accelerate the fluids. Secondly, it results from dynamic losses occurring where conduits change shape or direction. These changes cause turbulence in the fluid and loss of energy. To calculate these losses, use the formula:

$$H_e = \text{Loss} = F \times P_v \qquad \text{See Appendix 1}$$

where F = loss factor $\quad F = \frac{1-C_e^2}{C_e^2}$

P_v = (velocity (m/s)/1.29)^2

Table 1 lists the correction factors for nonstandard air.

Fluid flow in conveying systems is always dynamic, so expect pressure drops. Refer to the empirical formulas shown in the sample problems in Section 8 to calculate pressure drops. In addition, use the charts available (Figs. 7 and 8) to improve the accuracy of the calculations and to deduce the unknown parameters, such as surface roughness or material friction. Supplement the theoretical calculations with good judgment.

3.7 Air movers

Fans or blowers produce the air flows needed to circulate, exhaust, or deliver large volumes of air or gas. Two basic types of air movers are available: fans and positive displacement blowers.

Don't confuse the terms blower and fan. Both devices depend on centrifugal force to move air. However, blowers (or compressors) use positive displacement to draw air into a closed cavity, compress it, and release the pressurized air to the piping system. Fans, on the other hand, simply move air. A fan may be called a blower when the resistance to flow is predominantly downstream of the fan.

There are three main types of fans:

- axial fans
- centrifugal fans
- special fans

3.8 *Axial flow fans* Axial fans, also called propeller fans, provide high air flow rates against low static pressures. They are most commonly used for general ventilation or dilution applications. With these fans, a small change in pressure causes a large change in air flow rate. Fig. 9 illustrates three types of axial flow fans and provides typical performance curves for each.

These fans generally fall into one of the following categories:

- disc fans, moving clean air with no duct resistance
- narrow or propeller-type blade fans, used to move air against low static pressure, such as the air exhaust from buildings
- tube axial fans, built into short sections of ducts to move air containing material that could build up on the blades. Select large fans running at slow speeds for tube axial fan applications because less material collects on the blades, as compared with small fans operating at high speeds
- vane axial fans which offer power and space economy because they develop higher pressure than propeller fans and are available in multiple impeller and variable pitch designs. Use vane axial fans, however, for moving clean air only

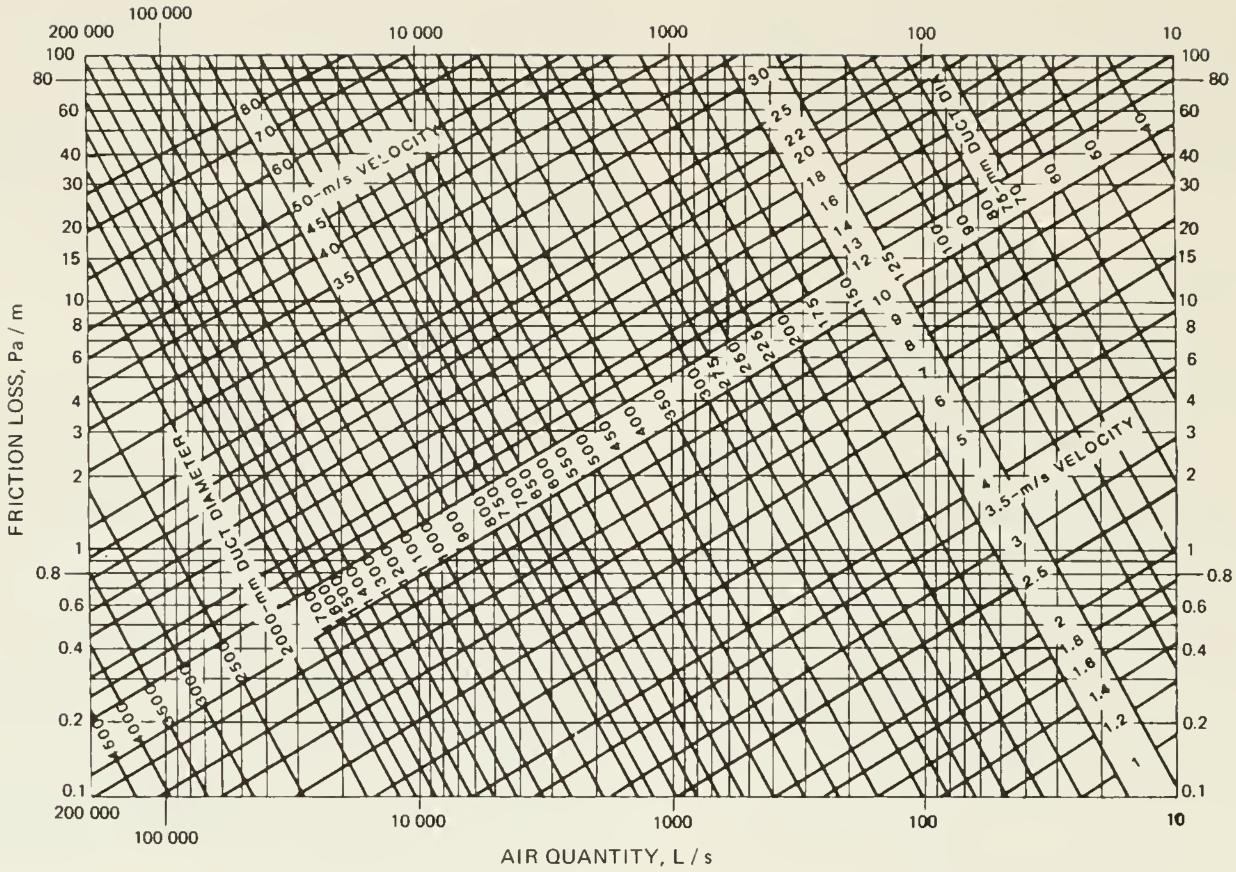


Fig. 7. Friction loss chart.
Source: ASHRAE handbook, S.I. edition.

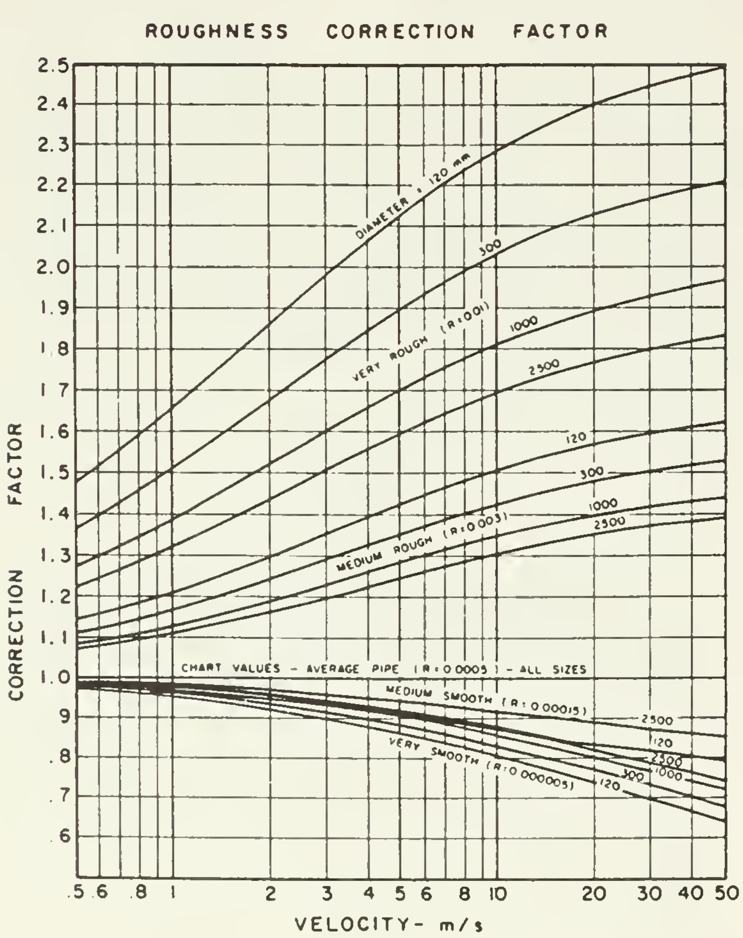


Fig. 8. Roughness correction factors.
Source: Industrial ventilation, a manual of recommended practice.

3.9 *Centrifugal fans* The centrifugal fan consists of a paddle wheel impeller rotating in a scroll-shaped casing. Fig. 10 illustrates the three main types of centrifugal fans — forward-curved blade fans, straight or radial blade fans, and backward blade fans — and provides typical performance curves for each.

The forward-curved blade fan has a multi-bladed, 'squirrel-caged' impeller with numerous small blades. The leading edge of each blade curves forward in the direction of rotation. This kind of fan works best at low static pressure (less than 1 kPa) and operates quietly. Use forward-curved blade fans against the low-to-moderate static pressures encountered in heating and ventilating systems. These fans are useful, for example, in grain drying applications where the operating pressure remains within a narrow range around 0.75 kPa.

Forward-curved blade fans are not suitable to move air containing dust or fumes. They are also inadequate in applications where the system design permits a change in air flow rate without a significant change in operating pressure. As static pressure increases, the fan may become unstable and its operation variable. Pulsing commonly occurs in such overload conditions.

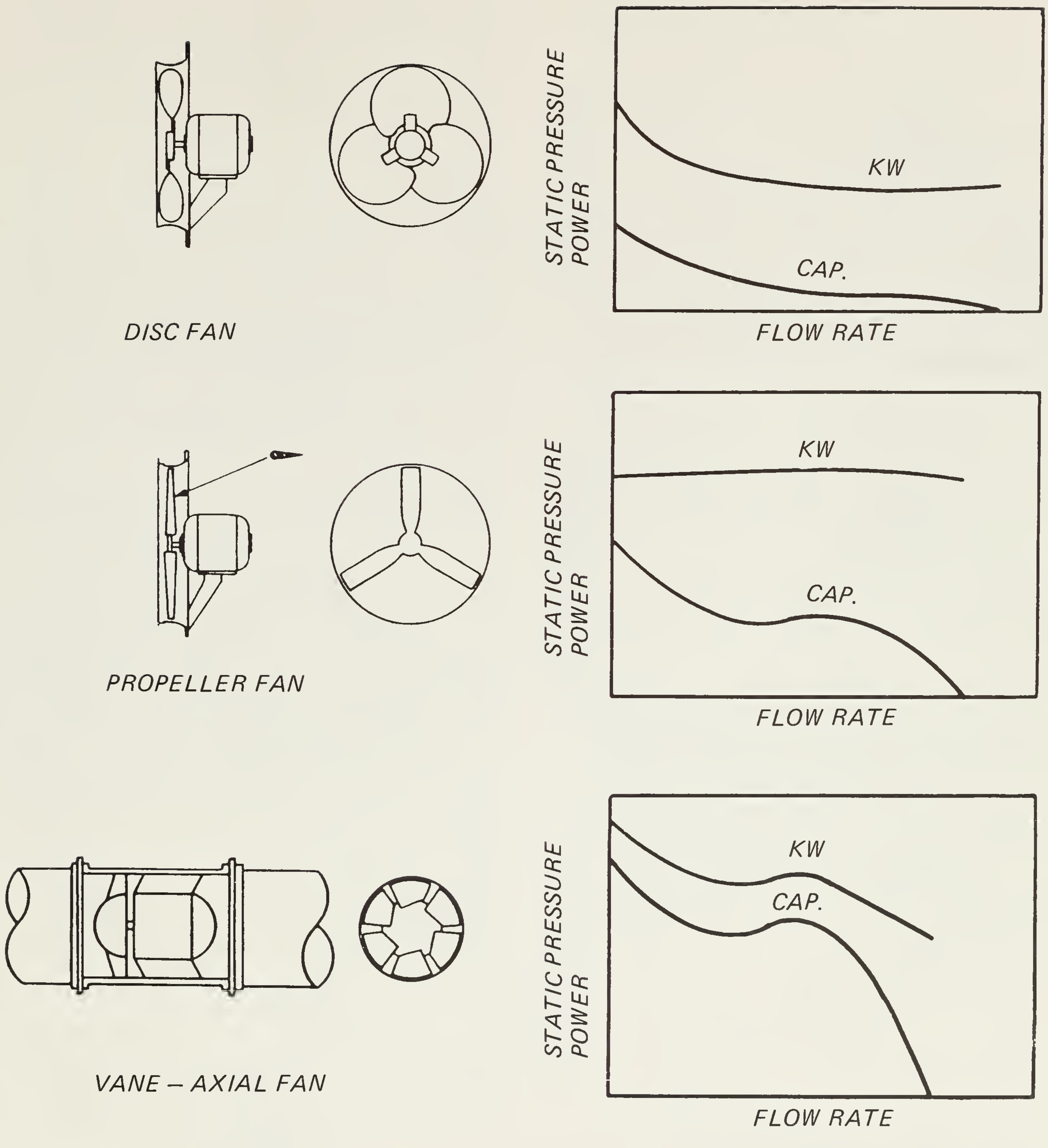


Fig. 9. Axial flow fans.
Source: Industrial ventilation, a manual of recommended practice.

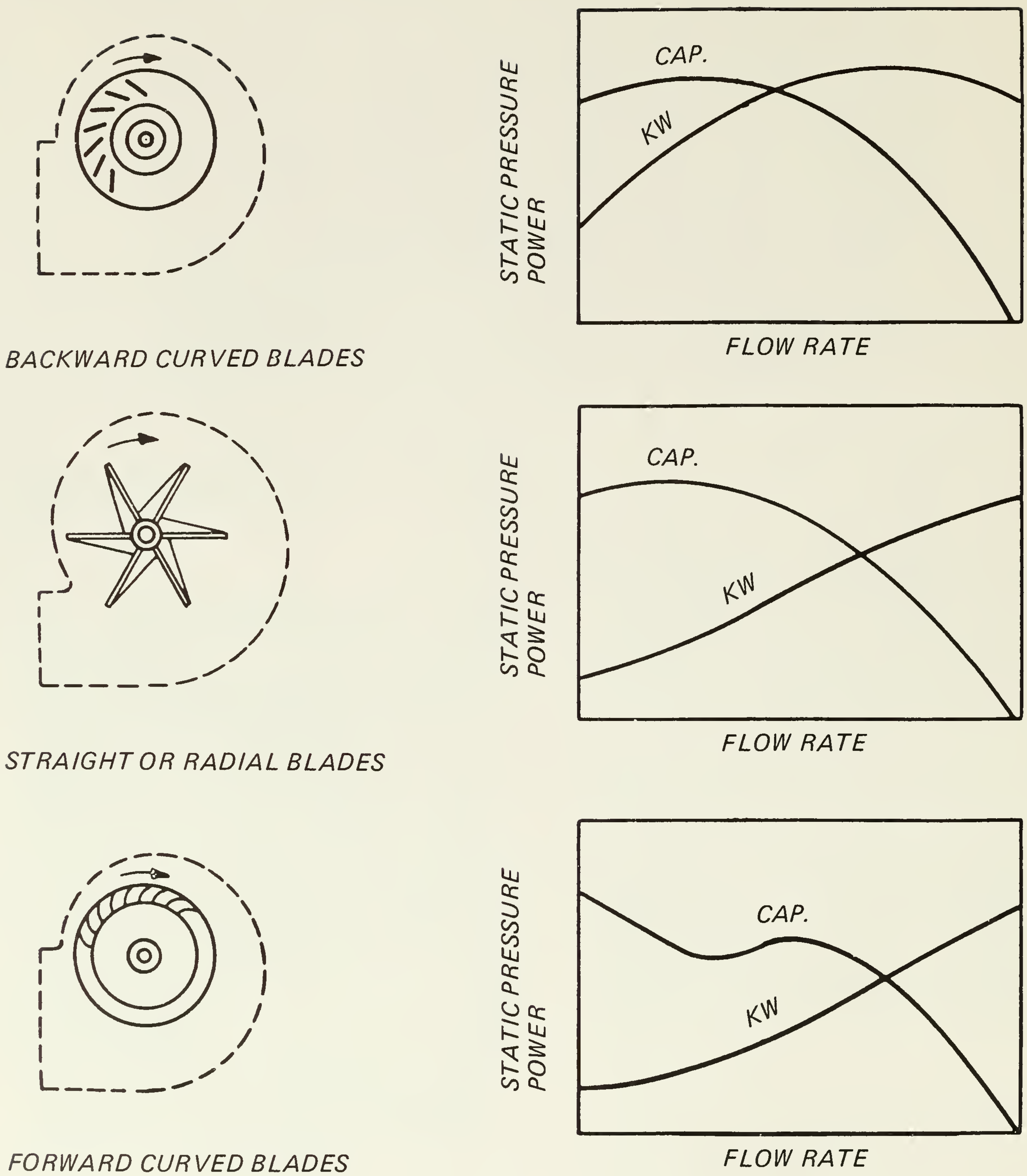


Fig. 10. Centrifugal fans.
Source: Industrial ventilation, a manual of recommended practice.

Straight or radial blade fans are the 'work horses' for materials handling and are suitable, with proper selection, for many applications. Use them particularly for applications where material passes through the fan or where high pressure is required.

Straight or radial blade fans operate at medium tip speed and noise levels. The wheel may be single or double shrouded or may have no wheel shroud at all, and various blade designs are available.

Backward blade fans rely on blades inclined backwards opposing the direction of rotation. The fan operates with a high tip speed, has a high mechanical efficiency, and resists overloading. This last characteristic is an important factor in conditions where a small change in static pressure causes a large change in air flow rate. In most agricultural applications (other than ventilation), however, overloading the fan is improbable because the system itself controls air flow rates.

Use backward blade fans for clean air only.

3.10 *Special fan types* Fig. 11 illustrates two special fans commonly used in agricultural applications:

- the airfoil backward-curved centrifugal fan
- the in-line flow centrifugal fan

The airfoil backward-curved centrifugal fan represents an advancement of the backward blade fan. The blades of this special type of fan are airfoil shaped in cross section, a change that creates a fan with higher efficiency and quieter operation. The airfoil backward-curved centrifugal fan also runs smoothly throughout its entire operating range.

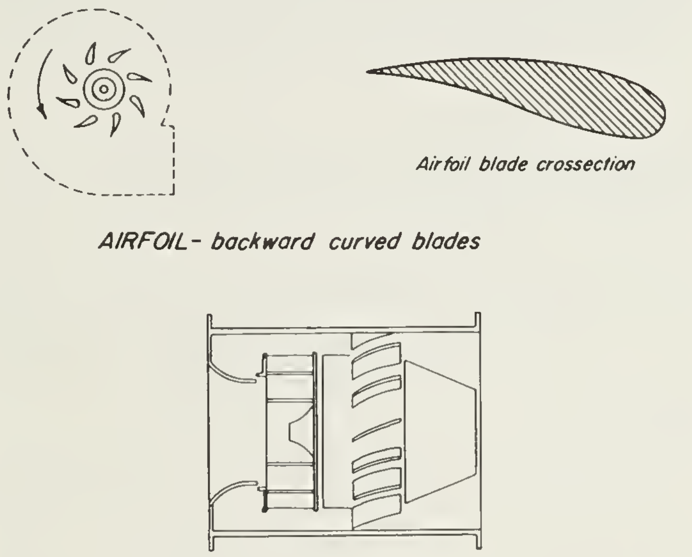


Fig. 11. Airfoil and inline centrifugal fans.
Source: *Industrial ventilation, a manual of recommended practice.*

In-line flow centrifugal fans are essentially backward-curved fans housed for installation in duct systems. Space requirements are similar to those for vane axial fans.

3.11 Positive displacement blowers and compressors

Operating pressure distinguishes positive displacement blowers and compressors.

Blowers are single-stage machines that can provide pressures up to 100 kPa. Compressors supply low volumes of air at high pressure.

The most common type of positive displacement blowers used for materials handling is the Roots two-lobe blower (Fig. 12) consisting of two meshing impellers geared together to form a progressive cavity. Air is drawn into the cavity, compressed, and discharged to the piping system. Because of the close tolerance design of these blowers, inlet air should be well filtered to avoid dust entry. Air containing dust severely reduces the life expectancy of these devices. As well, fit the system with a pressure-limiting device to protect against damage caused by pipe blockage.

Other types of low-to-medium pressure blowers, similar in design to the Roots version, include multiple lobe and screw lobe blowers.

There are two main types of compressors: reciprocating and rotary (Figs. 13 and 14). Individuals working in agriculture overwhelmingly favor reciprocating compressors for their simplicity, reliability, and low cost.

Typical performance for a 0.75-kW, single-stage piston compressor is 1.8 L/s at 700 kPa; a 7.5-kW unit can supply 23 L/s at 1200 kPa.

Table 2 lists capacities for compressors under continuous use.

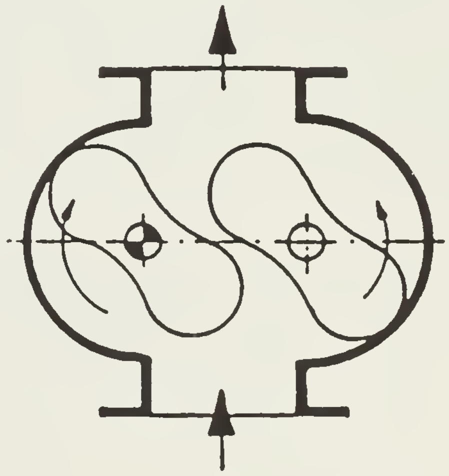


Fig. 12. Roots type blower.

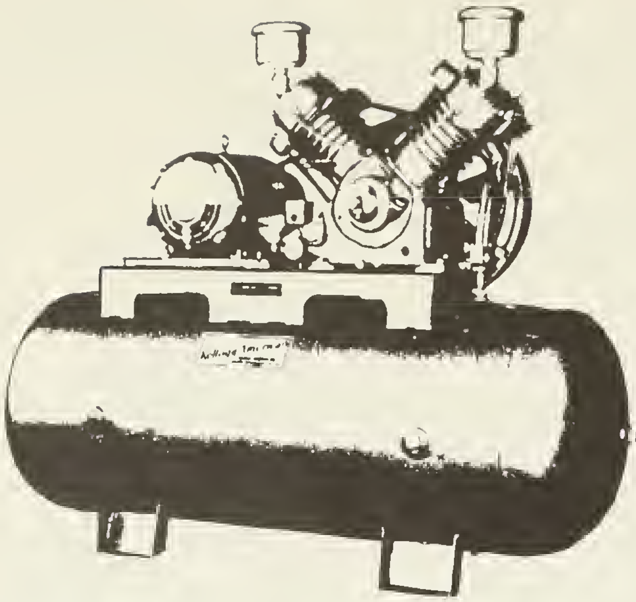


Fig. 13. Reciprocating compressor.
Source: Industrial catalogue: Air compressors.

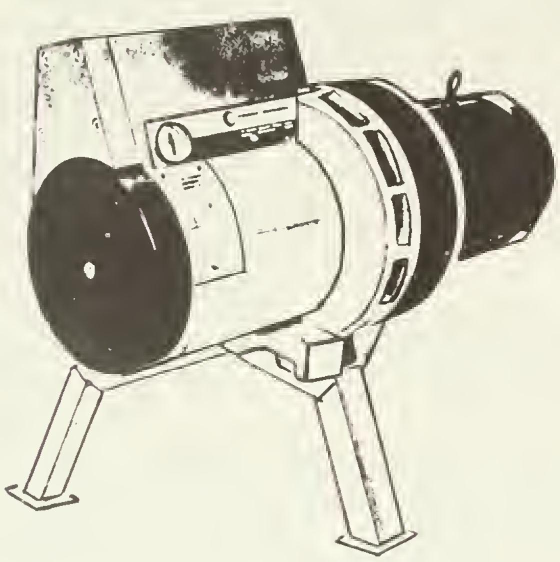


Fig. 14. Rotary compressor.
Source: Industrial catalogue: Air compressors.

Compressors of different sizes are available in a variety of configurations. Select the motor, receiver, and control combination based on the expected function in a given agricultural operation. Standard equipment necessary on all compressor installations includes a pressure switch for start-and-stop operation, safety valve, pressure gauge, receiver drain valve, nonreturn valve, intake filter, and safety shield.

Choose small receivers of volumes less than 200 L when storage capacity is not important, when demand is infrequent or small compared with the rated compressor delivery, or when constant speed controls are used. Select receivers larger than 200 L when the application requires large flows of short duration and minimum pressure variations, along with dry, cool air. In general, units with small receivers cost less for each power increment.

Table 2 Compressor capacities for continuous use

Pressure range, kPa	Consumption, L/s	Power, kW	
		Two-stage	One-stage
550–1030	1.5–1.8		0.75
550–1030	1.9–2.7		1.10
550–1030	2.8–3.6		1.50
550–1030	3.7–4.8		2.20
550–690	6.3–9.4	3.7	
550–690	9.5–13.8	5.6	
550–690	13.8–18.9	7.5	
550–690	18.9–28.3	11.2	
550–690	28.4–37.8	14.9	

Note: Capacities shown are for constant run operation. With intermittent usage an oversized tank allows the use of a lighter compressor.
Source: *Industrial catalogue: Air compressors.*

The low pressure range of most compressors is around 700 kPa. Industrial systems normally operate at 550–700 kPa. Set compressor pressures at 550–700 kPa on start-and-stop controls and 550–600 kPa on constant-speed controls. These pressures yield maximum delivery at minimum power consumption for each compressor size.

Use pressures around 850 kPa when there is heavy use of air at distant points and to increase the stored volume of air. Increasing the pressure from 700 kPa reduces slightly the rate of air delivery.

Agricultural operations rarely require high pressures of 1000–1200 kPa. These pressures are found mostly in commercial or industrial applications such as vehicle service shops.

4 PNEUMATIC CONVEYORS

Pneumatic conveying systems consist of a fan or blower to supply air, a feeder to introduce the material into the airstream, ductwork, and a separator to remove solids from the airstream.

Air conveys the material to its destination, providing the energy to accelerate the material from rest to the required velocity. At the required velocity, the fan or blower supplies the energy and air flow rate necessary to overcome the system resistance.

Properly designed pneumatic systems provide several advantages over mechanical conveying systems.

- They are self-cleaning.
- Pipelines can be run through difficult locations, though designers should limit the number of bends.
- They involve low maintenance costs (except for systems conveying abrasive materials).
- Dust problems are minimal.
- Some cooling and drying operations can be designed into the system.

Be aware, however, of these inherent disadvantages.

- Friable materials can be damaged.
- Pneumatic systems require more power than other forms of conveying.
- They are not suitable for conveying materials downward.
- Piping should be horizontal or vertical.

4.1 Types of pneumatic conveying systems

Pneumatic conveying depends on either a negative or positive pressure. Negative-pressure systems operate below atmospheric pressure; positive-pressure systems operate above atmospheric pressure. Fig. 15 illustrates both types of configurations.

Select a negative-pressure system where there are several pickup points but a single discharge. Conversely, choose a positive-pressure system when there are several discharges and a single pickup.

In either arrangement, the air flow rate necessary to convey the material determines the power required. The pressure and air flow needed for the system establishes the fan or blower requirements.

In some applications, pneumatic conveyor systems maintain a permanent location within a plant. In other cases, the conveying system can be moved to various locations. Mobile pneumatic conveyors typically transfer materials from one point to another, for example, unloading grain from a bin to a truck or from a rail car to a bin or truck.

Pneumatic conveying systems are classified by the operating pressure and fall into four categories:

- low-pressure systems
- medium-pressure systems
- high-pressure systems
- fluidizing systems

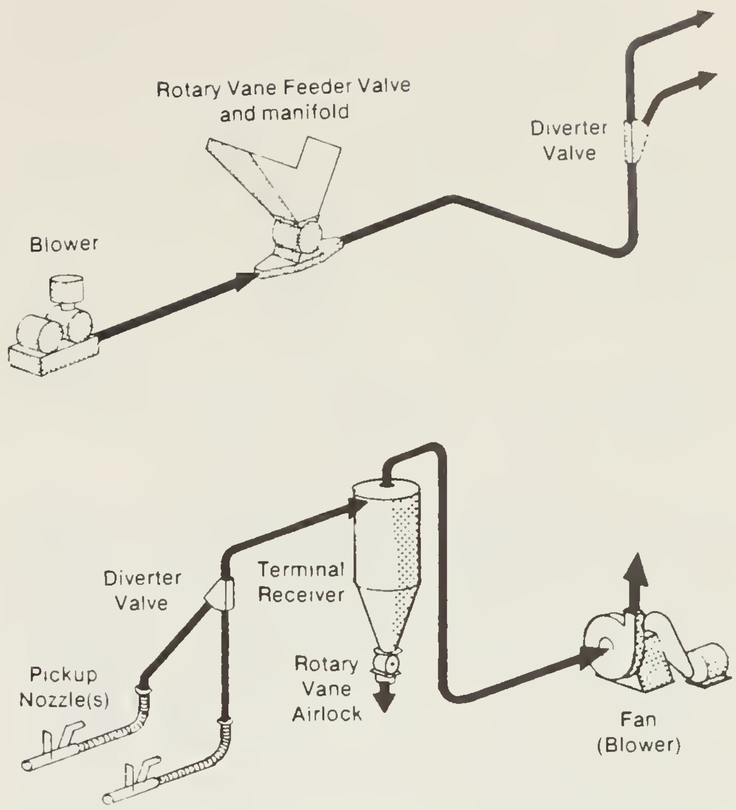


Fig. 15. Positive- and negative-pressure pneumatic conveyors.
Source: ***Industrial catalogue*****, Muncy: Koppers Company, Inc.**

4.2 *Low-pressure systems* Low-pressure conveying systems operate at pressures up to 5 kPa with solids-to-air ratios of 1.0 or less. A centrifugal fan supplies air.

These are usually negative-pressure systems, useful for conveying light, dry materials, such as dust, bran, or mill feeds. With such fine materials the mixtures can pass through the fan with no danger of them degrading.

In low-pressure systems, ignore the presence of suspended solids if the solids-to-air ratio is less than 0.2. (The solids-to-air ratio equals the mass flow rate of material being conveyed divided by the mass flow rate of air.)

At high concentrations, the presence of suspended solids requires consideration because particle acceleration and the friction of material on the duct walls increase the power requirements for the system. For more information on their impact on the design of conveying systems, see Section 4.13, "Conveying capacity," and Section 8, "Sample problems."

4.3 *Medium-pressure systems* Medium-pressure systems, which can be either positive or negative, operate at pressures of 5–35 kPa and with solids-to-air ratios of 1.0–6.0. Blowers, either single- or multiple-stage or positive displacement, supply air. System capacities normally range from 1 to 100 t/h. Larger systems are available for facilities that unload ships.

Use medium-pressure pneumatic conveyors, either positive- or negative-pressure systems, to move granular materials or pellets over short distances in small diameter ducts.

4.4 *High-pressure systems* These systems operate at pressures of 35–170 kPa, with solids-to-air ratios up to 30. Solids-to-air ratios this high can cause solids to fluidize because of air entrainment.

Use high-pressure systems to convey granular materials at high flow rates over long distances.

4.5 *Fluidizing systems* Fluidizing systems operate at low air line velocities and ratios of solids to air up to 300. Use these systems to convey finely divided solids capable of entraining air (e.g., starch, flour, or fine lime). Fluidizing systems may be either blow-tank or airslide conveyors (e.g., Airslide, TM Fuller GATX).

In a blow-tank system, the material is first fluidized with air and the tank then pressurized. Finally the material is discharged through a pipeline to storage. Use blow-tank systems to elevate materials to storage. They are suitable for conveying horizontally only over short distances.

Airslide conveyors consist of a formed, closed casing mounted on a slight negative incline. A fabric or other porous membrane horizontally separates the casing from the conveyor system. The lower chamber, or plenum, is pressurized with air from 6 to 30 kPa. This action fluidizes the fine material and causes it to flow. Both the air flow rate and pressure required to operate airslide systems depend on the characteristics of the material transported and the desired material flow rate.

Do not use airslide systems to elevate materials. The conveyors are installed in these systems with a downward slope in the direction of material flow.

4.6 Feeders

In pneumatic conveying systems, feeders introduce material into and separate it from the airstream.

In low-pressure systems, a suction nozzle (a venturi) or a plugged-screw feeder directs material into the airstream. Ensure that sufficient bleed air can enter the system at the material pickup point to maintain a low material-to-air ratio. Without enough bleed air, the system will plug.

In medium- and high-pressure systems, because of the pressures involved, use a minimum-leakage feeder. Rotary airlock feeders are usually suitable, although adjustable bleed intakes are common on high-capacity systems that combine negative intake and positive discharge.

4.7 Rotary valve feeders

Use a rotary valve or airlock (Fig. 16) to introduce material into the airstream in both positive and negative systems. Rotary valves can also function as air seals at the material discharge point of collectors. Fig. 17 illustrates a blow-through air lock and feeder.

In operation, material flows into the top of the air lock, the intake, and is carried between the vanes around the horizontal shaft to discharge out the bottom. Gravity carries the material through the system.

Three factors dictate the capacity of the air lock:

- displacement of the air lock
- rotation speed of the rotor
- ability of the material to flow into and out of the rotor

The displacement is readily calculated for an air lock. Multiply displacement by the speed of rotation to calculate the theoretical capacity of the air lock.

The ability of material to flow into and out of the air lock is much less predictable. Keep the centrifugal force on the material less than the gravitational force to ensure that the material falls into a vane space. This limitation generally restricts the rotational speed of air locks to 20–60 r/min, depending on the diameter of the rotor, the ability of the material to flow, and the expected life of the device.

The pressure differential across the air lock can also reduce the capacity of an air lock. If the difference in the direction of material flow is positive, little if any effect results. However, where a negative pressure differential exists, three factors can reduce capacity:

- prepressurization of the return interstices. The interstices must vent through the material at the intake before the material can flow into the pocket.
- leakage of air through the air lock from the discharge side to the feed side. This leakage can impede the flow of material into the air lock.

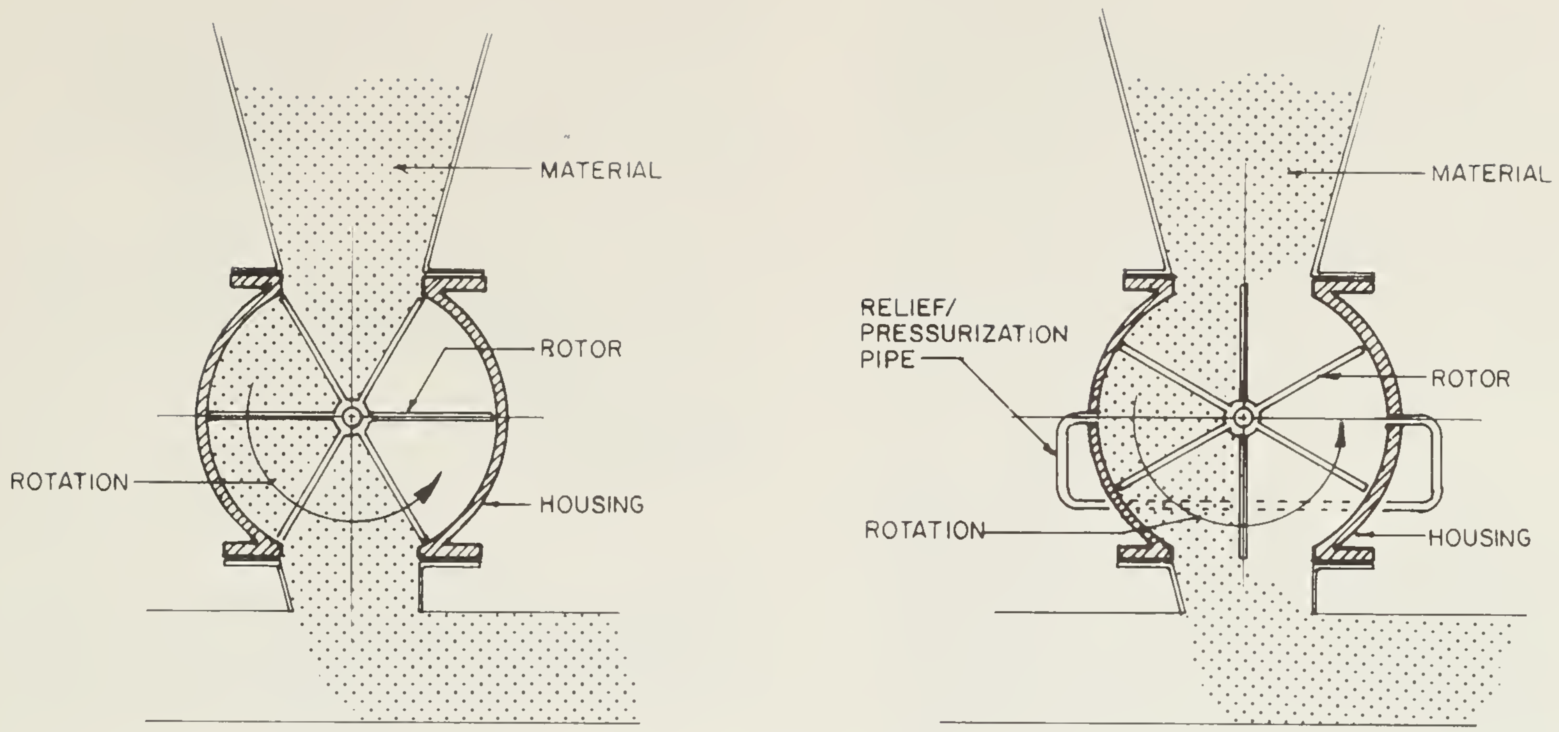


Fig. 16. Rotary air flow regulator and high-pressure rotary feeder.

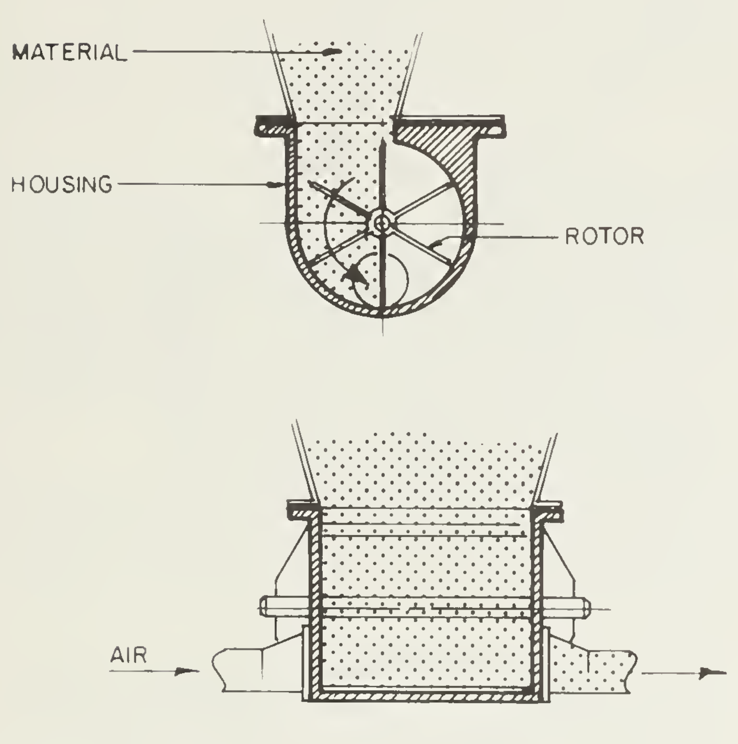


Fig. 17. Blow-through airlock and feeder.

- pressurization of the discharge duct. Air pressure here can prevent the material from discharging between the vanes.

Venting the return pockets to atmospheric pressure through the casing can control prepressurization. Create the vent at a point where the pocket separates from both the supply and discharge ports by a distance equal to the space between two vane tips.

Control air leakage by selecting a close tolerance rotor or by adjusting the tip-to-casing clearance. Clearance adjustments are possible on adjustable tip or flexible tip air locks only.

Selecting a blow-through type of air lock or prepressurizing the supply vanes with air vented from between the return vanes controls pressurization of the discharge duct.

Inadequate time allowed for material to flow into or out of the vanes further reduces the capacity of a rotary air lock. This time factor is integral with the presence or absence of backwards air flow through the air lock and the centrifugal force, both discussed earlier.

The ability of material to flow also affects capacity. Sticky or rough material flows slower than dry granular material. For example, a system handling smooth wheat kernels does so at a greater capacity than another handling rough barley kernels, despite air locks of equal size, speed, and air pressure differentials.

Adjustable-tip and flexible-tip rotors are the most common air locks. Both these systems generally provide a clearance of 0.07–0.25 mm between the rotor tip and casing. Flexible-tip units require about three times the power for operation at –30°C compared with more temperate conditions.

Wear on the tips of the vanes causes air to leak backwards through the valve. This situation not only impairs the capacity of the valve but may compromise the performance of the entire pneumatic system. Adjustable tips maintain tight clearances between the rotor and the shell for minimum leakage. Where the pressure differential across the valve permits their use, flexible-tip vanes prevent leakage better than other types of vanes. The longevity of flexible-tip vanes is less than fixed-tip, but this poses little problem in most agricultural applications.

Select the number of air lock vanes depending on applications. When designing a feeder system, the number of vanes must yield the most uniform rate of air and material mixing. When planning a system to discharge a pressurized container into another at atmospheric pressure, select the number of vanes on the basis of acceptable leakage. Five or six vanes are generally best in this case so four vanes are always in contact with the shell.

Medium-pressure feeders require six or, more commonly, eight vanes to vent and pressurize the interstices.

The number of vanes dictates the volume of the interstitial space for any air lock. Large quantities of material discharged into an airstream cause air velocity to rapidly decrease and static pressure to increase. These changes in operating conditions each time a vane passes by the air lock discharge port can result in system surges. Surging increases power requirements, decreases conveying capacity, and causes premature wear of the drive arrangement. Roots blowers (see Section 3.11) are also prone to surging. When using them near an air lock, provide Roots blowers with a surge tank to reduce pulsation.

Each manufacturer supplies data on the rotor displacements of their products. A sample of such data for Type 1 Sprout-Waldron rotary valves follows.

Typical volumetric displacement

Type 1 Sprout-Waldron rotary valve

Diameter to length (mm/mm)	200/150	250/200	360/250	400/360
Capacity (m^3/r)	0.003	0.008	0.020	0.037

Source: *Industrial catalogue*, Muncy: Koppers Company, Inc.

Select the capacity based on 75% of displacement shown in the table above. And for a system expected to require minimal maintenance, set the maximum speed at 30–40 r/min. Drive power depends on the system manufacturer, but is usually rated at 0.2–1.2 kW. A high-pressure differential across the feeder increases both the power requirement and the need for close tolerance between the rotor and casing.

4.8 *Sample calculations for rotary valve speed* Calculate the necessary valve speed for rotary valves required to feed material with density of 800 kg/m^3 at a rate of 50 t/h.

$$\text{Volume/min} = \frac{50}{0.80 \times 60} = 1.04\ \text{m}^3/\text{min}$$

Displacement required at 25 r/min

$$= \frac{1.04}{0.75 \times 25} = 0.056\ \text{m}^3/\text{r}$$

Displacement for a 400/360 valve for one revolution

$$= 0.037\ \text{m}^3/\text{r}$$

Revised rotary valve speed

$$= \frac{0.056 \times 25}{0.037} = 38\ \text{r/min}$$

4.9 *Choked screw feeders* Use choked screw feeders (Fig. 18) to move material from a bin to a conveying pipe. The material acts as an air seal, but the hopper must have a deep column of grain above the feeder to prevent air leakage or flushing. This system is particularly suitable for low-pressure systems.

Screw feeders transport material to pneumatic conveyor systems. The conveyor discharges directly into a hopper on the pneumatic conveyor.

4.10 *Injector feeders* Injector feeders (Fig. 19) have little application in the agricultural industry. They are useful for accelerating particles with a high specific density. Unfortunately, the negative pressure developed in the venturi throat causes flushing with subsequent loss of control over the conveyed material.

4.11 *Suction nozzle feeders* Suction nozzle feeders (Fig. 20) accept material from low- or medium-pressure systems. Secondary air enters the system through a flared opening or via air bypass. When designing a system with a suction nozzle feeder, consider controlling the bypass air to optimize the material conveying rate.

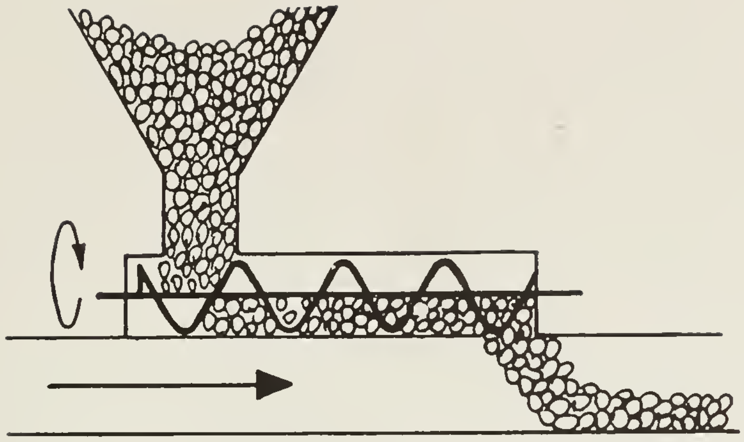


Fig. 18. Choked screw feeder.

Suction nozzle feeders can be used with a hose, like a vacuum system, or they can unload from a pit designed to allow air into a fully-loaded system.

4.12 Particle and air velocity design factors

Successful design of a low-pressure conveyor system depends most on the air velocity. The air velocity, however, is difficult to establish since it varies with the characteristics of the conveyed materials.

When designing pneumatic conveying systems, consider these characteristics of the materials:

- ability to flow
- density
- particle shape and size
- abrasiveness
- tendency to damage

The ability to flow is probably the most important design factor. Dry, granular materials usually flow smoothly and present no problem. Oily or sticky materials, on the

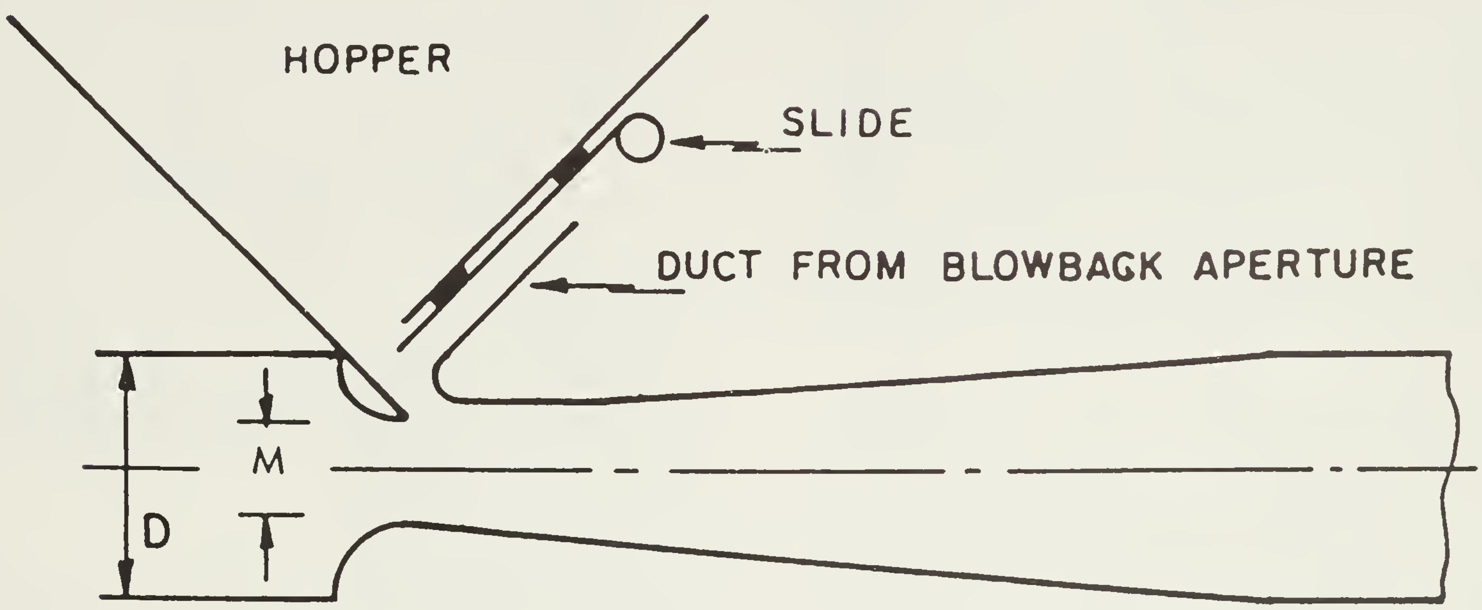


Fig. 19. Injector feeder.

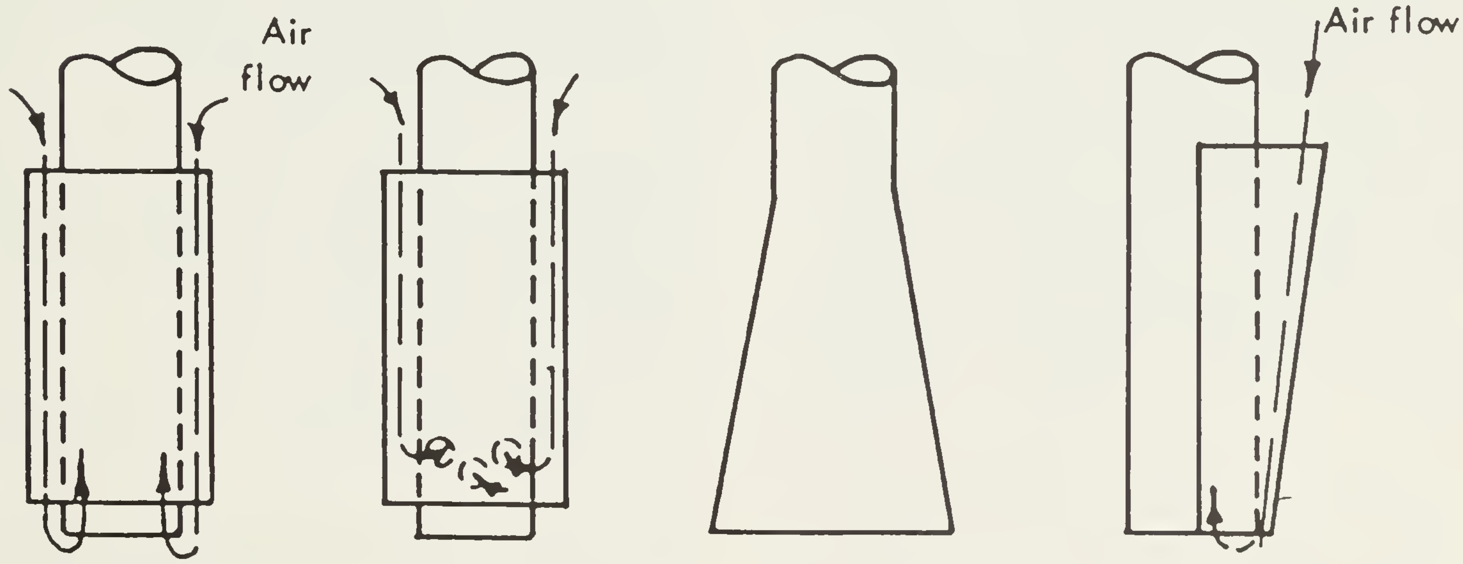


Fig. 20. Suction nozzle feeders.

other hand, can build up on the pipe walls and plug the conduits. As a result, conveying systems transporting these materials require large power supplies. The moisture content of materials conveyed affects both their ability to flow and their cohesiveness.

Free-flowing materials with a bulk density of 400–1400 kg/m^3 do not usually present problems in conveying. Denser materials require more power to move them; lighter materials require larger feeders.

Spherical particles up to 10 mm in diameter generally convey smoothly, with no special requirements. Whereas wheat and soybeans have approximately the same specific density, the larger spherical soybeans require 15–20% greater velocity than does wheat for conveying.

Some agricultural products are fairly abrasive, for example barley, rice, soybeans, and granular fertilizers. Systems conveying these materials require more power to overcome the friction and drag they cause. As well, component wear can be excessive in systems conveying abrasive material.

Pneumatic conveying systems may cause damage to some agricultural materials. Seed grain and oilseeds are especially susceptible. Meal and groundstock, on the other hand, transport with little or no deterioration. Careful system design and equipment selection can minimize potential damage caused by the conveying system.

4.13 Conveying capacity

Besides the influence of air flow rate and velocity on conveying capacity, pneumatic conveying systems also depend on steady feed rates to operate optimally. Intermittent feeding reduces the system capacity and may cause pipe blockage.

In a low-pressure, negative system, introduce secondary air at the material pickup point to optimize conveying capacity. The quantity of secondary air depends on the conveying rate and the material-to-air ratio required. Lowering the secondary air intake rate increases the material pickup rate; however, plugging occurs if the secondary air intake rate is too low. Unfortunately, the conveying rate at which blockage occurs is virtually impossible to estimate accurately. Table 3 lists some commonly accepted conveying velocities.

In general, the material and the solids-to-air ratio govern conveying velocity. For materials not listed in Table 3, and at densities of 400–1400 kg/m^3, use Table 4.

Table 3 Typical conveying velocities

Material	Air velocity, m/s
Beans	20–25
Barley	20–33
Corn	25–30
Cotton seed	20–30
Oats	20–30
Sand (dry)	28–46
Sawdust	20–30
Shavings	18–25
Wheat	20–30
Wool	23–30

Table 4 Conveying air velocity for various solids-to-weight ratios

Weight ratio : Solids/Air	Velocity, m/s
0.33–0.13	20
0.44–0.33	21
0.53–0.44	23
0.67–0.53	25
0.85–0.67	26
1.11–0.85	28

Source: *Reference data and tables. Bulletin RD-1.*

4.14 Separators

Separators remove the solids from the airstream. Of the various types of separators available, two are used extensively in agricultural applications: the cyclone and the fabric collector or filter.

4.15 *Cyclone separators* The cyclone is a centrifugal separator. A medium efficiency air cleaner, it performs best on relatively coarse, particulate matter. Cyclone separators are generally not suitable for fine dusts (particles of less than 50 µm), although separation efficiency varies dramatically with the cyclone design and condition of maintenance.

In a cyclone separator, the air and solids are forced to spiral down the conical chamber. The centrifugal force causes the solids to move against the outer wall of the cyclone. Solids are removed out the bottom of the cyclone while the air discharges through the top.

As the cyclone diameter decreases and air velocity increases, the pressure drop increases and separation efficiency of the system increases.

Separation efficiency depends on inlet velocity, and cyclone diameter and length. High-efficiency cyclones have high inlet velocities

and small diameters, and are long. They also create large pressure drops.

For agricultural systems, choose low-pressure-drop cyclone separators of large diameter (Fig. 21).

When a system discharges only clean air, use high-efficiency cyclone separators that are tall and narrow with a high inlet velocity. Alternatively filters are useful.

Pressure drop through cyclone separators varies from 2 to 5 times the inlet duct pressure. Provide the cyclone with an air seal valve on the discharge outlet. Set hopper cone angles at 60° or steeper to prevent material build up.

Many jurisdictions now regulate the opacity of discharge air. This situation requires the use of air filtration equipment to remove the fine particles in grain dust. Often more than 50% of dust by weight is less than 50 μm in size.

In the design of cyclone or filter systems, provide access hatches so cleaning is possible. Expect the system to plug occasionally, particularly in feed handling applications. Dust containing moisture or fat is especially prone to build up on the inside of the cyclone and must be cleaned off. Failure to clean the separator reduces its separation efficiency and increases the risk of product bridging.

To ensure efficient removal of particulates from the air, maintain the direction of air rotation in the cyclone by a tangential discharge of air in the same rotational direction as the intake. Be careful that fans do not disturb the rotational direction of the air flow in the system.

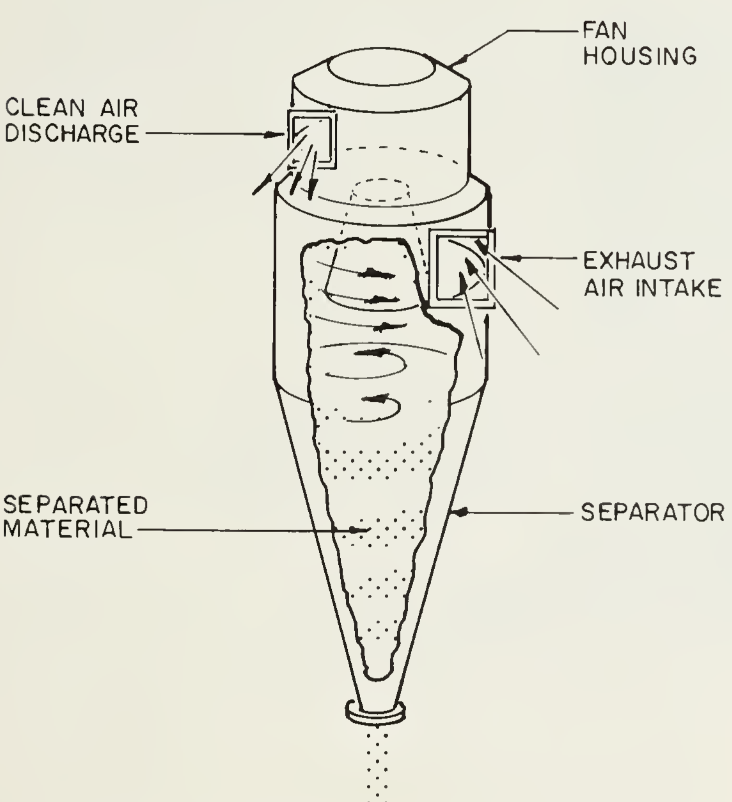


Fig. 21. Cyclone separator.

Cyclone feeder size depends on the air flow rate and the material to be handled. Fig. 22 gives typical dimensions for large diameter cyclone separators. Many companies manufacture cyclone systems; consult their product literature for more detail.

4.16 *Fabric collectors* Fabric collectors efficiently remove fine particulate matter from the airstream. They consist of a number of fabric bags or tubes in an enclosure.

As dirty or dusty air passes through the fabric, it deposits the dust on the fabric. The clean air is then recovered or discharged. Separation efficiency is commonly 99% for particles of 5 μm and larger.

Mechanical shaking or a reverse flow of clean air dislodges dust from the filter bags.

Two factors dictate fabric collector size: the amount of air passing through the fabric, and the ratio of air to cloth (also called the ratio of the air flow rate to filter media area) for the material handled.

Air-to-cloth ratios vary from 15 (L/s)/m^2 for mechanical shakers to 75 (L/s)/m^2 for air cleaners, depending on the material handled. Mechanical shakers are not particularly effective for most agricultural materials. Use instead filters equipped with reverse air jets

L/S	A	B	C	D	E	F	G	H
700	190	50	200	610	50	400	300	760
1100	240	65	250	710	75	460	355	860
1300	250	75	300	760	100	510	380	910
1700	280	90	355	910	150	585	460	1170
2600	330	115	405	1070	150	685	530	1320
3400	330	150	460	1170	150	740	585	1420
4000	380	150	510	1270	250	890	710	1525
5500	430	190	610	1420	250	915	710	1575
7000	480	215	710	1625	250	1040	810	1625
9500	610	230	810	1830	300	1145	890	1830
12100	760	230	910	2030	300	1295	1015	2030
14200	840	255	1015	2235	300	1320	1120	2235

Dimension I – Telescoping Inlet Sleeve for Adjustment

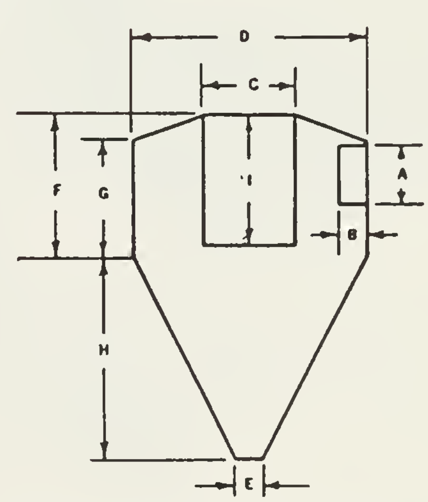


Fig. 22. Typical dimensions for large cyclones.

supplied by low pressure fans or compressed air. Filter pressure drop should measure 1.0–1.5 kPa.

Consult manufacturers of filter collectors for recommendations on fabric size and selection.

4.17 Conveyor system safety

Except in low-pressure systems, install pressure or vacuum relief valves in the conveying pipeline as close to the blower as practical. Set the pressure relief valves at 12–20 kPa above operating pressure. This relief valve protects the blower from damage caused by pipe blockage.

A positive-displacement blower requires a good quality inlet filter. Check the filter periodically and replace it when required. In addition, equip the blower with a discharge snubber. The snubber serves two functions. Firstly, it suppresses the surging that is inherent to Roots type blowers; secondly, it reduces noise levels.

Ideally, locate the blower in an unoccupied space. If the blower must be installed in an occupied area, enclose the blower in an acoustic cover. In either situation, mount the blowers on vibration damper pads.

As with all materials-handling equipment, keep belt guards and protective covers in place during operation.

5 FORAGE BLOWERS

Impeller blowers are commonly used to elevate silage, chopped hay, high-moisture grain, meal, and other materials that meter directly into the intake of the fan. The material then leaves the impeller housing at a velocity approaching the peripheral speed of the impeller blades. Initially, this velocity exceeds the air velocity, causing the air to have a retarding effect. As the material rises in the pipe, however, its speed decreases and the air imparts some energy to the material to convey it up the pipe. For this reason, although the addition of more blades to the impeller increases the air delivery, it has very little effect on particle velocity.

The housing for the impeller is usually concentric with little clearance between the impeller blades and the housing. Minimum clearance ensures maximum blower efficiency. Fig. 23 shows the components of a forage blower system.

5.1 Power requirements

Because of their low efficiency, forage blowers require higher specific power than do mechanical conveyors. The actual power levels required to operate a forage blower depend on fan speed, type of forage, and feeding rate and uniformity. As predicted by

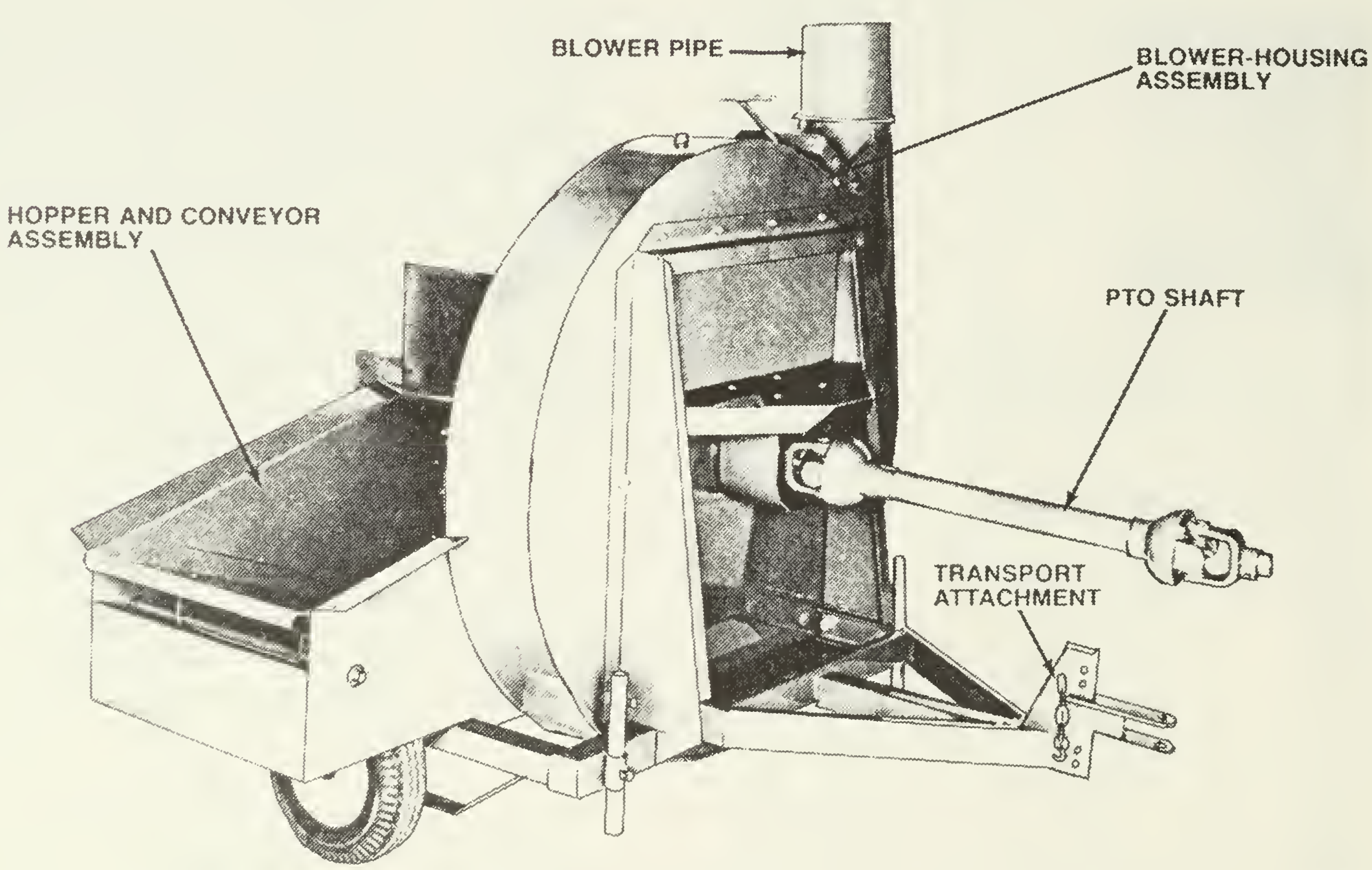


Fig. 23. Forage blower system.
Source: Evaluation report: John Deere 6500 forage blower.

the fan laws, the capacity of a forage blower varies linearly with speed whereas power varies exponentially with speed. Therefore, to obtain the most efficient operation of a forage blower, operate it at the lowest speed while maintaining the desired capacity and elevating height.

Typical average power requirements for forage blowers are 3–4.5 MJ/t for corn or grass silage and 3–6 MJ/t for hay.

Operational efficiency of radial-blade concentric blowers is usually 25–30%. Undesirable particle movement, primarily when particles are not released from the impeller before striking the cut off, generally accounts for this poor efficiency. For ideal operation, adjust the feed opening location and feed rate so the material slides off the blades and discharges up the pipe with a minimum of housing friction. No material should pass the cut off. However, in practice, these conditions are difficult to achieve.

5.2 Capacity of forage blowers

The theoretical height to which material can be elevated is:

$$H = V^2/2g$$

where H = elevating height (m)

V = peripheral velocity (m/s)

g = acceleration due to gravity (m/s^2)

However, friction in the discharge pipe and the energy required to carry the material around the discharge elbow result in heights considerably less than predicted.

Experiments show that actual elevating heights vary from 40 to 50% of theoretical values. Use the following equation to calculate practical conveying heights for pipes up to 250 mm in diameter.

$$H = \frac{3.1\,v^2}{gW^{0.33}}$$

where W = blower capacity (kg/h)

This equation is valid for capacities up to 5.5 t/h and accounts for the bend at the top of the pipe where material enters the silo.

In practice, blower capacity varies widely. Evaluations of two forage blowers by the Prairie Agricultural Machinery Institute (PAMI) reported the following performance. For corn at 60% moisture content, specific capacity ranged from 20 to 27 t/h. For clover at 32% moisture content, specific capacity ranged from 25 to 33 t/h. Both materials were elevated 25 m. The power takeoff speed was 540 r/min. A 40-kW tractor sufficiently operated the forage blowers. PAMI also found the most efficient power use occurred when material was conveyed at maximum conveying rates.

As a general rule, specify the capacity of the forage blower at twice the capacity of the forage harvester. This setup allows the blower to keep up with or ahead of the harvesting and transporting operations.

If peripheral speed is too low, blockage occurs. Operating at speeds greater than required uses excessive power.

Several systems can be used to convey feed to a forage blower, including augers, belt and chain conveyors, vibrating feeders, and suction systems. Feeders are available in lengths of 0.5–3.7 m. Tractor power takeoff usually drives forage blowers at either 540 or 100 r/min.

5.3 Impeller blowers for granular materials

Impeller blowers for handling grain may be classified as forage blowers running at reduced speeds, or impeller blowers designed to convey grain vertically.

Forage blowers handle grain in the same way as silage. Damage to the grain may be extreme, making forage blowers unsuitable for seed or commercial grain. They are useful, however, for feed grain and meal.

Impeller blowers are smaller than forage blowers and operate at higher speeds. Material usually feeds into the eye of the impeller, avoiding impact with the impeller blades and reducing damage.

5.4 Operating recommendations

Consider the following suggestions to maximize blower performance and ensure operator safety.

- Use solid elbows with large radii and locate them as close to the discharge end of the system as possible. This configuration reduces friction and power requirements.
- Set pipe as vertical as possible and avoid dents.
- Allow 200–250 mm clearance from the bottom of the system's discharge to the top of the silo. This configuration relieves any back pressure that can cause plugging.
- To obtain the greatest capacity, use the largest specified pipe and accessories for the blower.

- Add water to a running blower through the water inlet to dissolve gum deposits left by sticky silage. Check the fan between loads and adjust the amount of water as required.
- Ensure the blower is in a stable, level position when operating.

6 MATERIAL DAMAGE CONTROL

Very little can be done to reduce the damage caused to grain or pellets during pneumatic conveying. The high surface-to-weight ratio of particles of ground materials causes them to move in conveyors relatively easily and damage goes unnoticed. Grains, on the other hand, may receive up to 3 or 4% damage in a single handling through a pneumatic conveyor. In general, adherence to the following guidelines may reduce material damage, system wear, and power requirements.

- Design to minimize the number of changes in direction. Elbows are major wear points as the material rubs against their exterior faces.
- Use long-radius elbows that are cold formed in a single piece. Where ducts are too light to cold roll, as in dust-conveying systems, use at least seven segments in a 90° elbow. Systems conveying dust require elbow radii of at least 2.5 times the duct diameter (*D*). When conveying ground materials, use elbows with radii of at least 5*D*. Use elbows with radii of 10*D* for conveying grain or pelleted material through rolled bends.
- Avoid using rotary air lock feeders to convey pellets.
- Use the lowest conveying speed that yields continuous reliable conveying performance.
- Use abrasion-resistant liners on elbows subject to high wear. Liner materials include ceramics, high-carbon steel, and ultra-high molecular weight (UHMW) polyethylene.
- Some particularly abrasive materials cause pneumatic conveying systems to wear rapidly. Among these are peas, soybeans, flax, and grain corn. In the case of peas and soybeans, dirt frequently accounts for most of the wear.
- Some materials are particularly sensitive to abrasion and are unable to withstand high impact forces. These include the dicotyledenous seeds such as peas and beans; seed grain, where minor damage may reduce the germination rate; and improperly dried grain that may contain cracked or case-hardened kernels.

7 DUST CONTROL

Dust control is essential in facilities that handle agricultural materials. Dust poses a danger of fire or explosion and a health hazard.

When grains are conveyed, friction between kernels or between kernels and container walls creates dust. The dust is a very fine, organic substance with a high fuel value. As the proportion of dust in the air increases so does the danger of explosion. As well, research has shown that grain dust can lead to serious respiratory ailments in workers.

Dust control consists of two components: preventing dust release to the atmosphere, and collecting dust where it is generated.

7.1 Control of dust generation and release

The capital and operating costs for a dust collection system are extremely high, especially since the system generates essentially no income. In addition, annual maintenance costs can reach 4% of replacement cost and the addition of a dust collection system to an existing facility often doubles the power consumption and demand charges.

Much can be done in the design of conveying facilities to reduce the need for special dust collection equipment. Minimizing kernel friction is particularly important and easily controlled by nonturbulent flow systems. Other means to reduce friction in grain handling include:

- using chain or belt conveyors rather than screw conveyors.
- reducing spout slope and length to slow the grain speed in the spout. This design feature reduces spout wear, grain damage, and dust generation, both along the spout and at points where the material changes direction, for example, at the end of the spout where the grain drops into a bin or onto another conveyor.
- using spouts with abrasion-reducing dead ends and flow retarders.
- reducing bin depth to limit the impact that occurs as bins fill. More than half the grain put into a bin drops at least half the bin depth.
- reducing the turbulence of grain flowing through transitions between one piece of equipment and another. A transition designed to constrain the natural flow of product rather than to redirect it generates much less dust and requires much less maintenance.

- designing plug flow-control devices to prevent entrained dust from becoming airborne. This style of design requires several small surge hoppers but eliminates dust generation and release. Modern automatic flow-control devices help in this area.
- using large, slow conveyors to reduce abrasion and turbulence at both the feed and discharge points. Besides controlling dust, these conveyors also last longer and require less maintenance.

Attention to certain operating procedures can also control release into the workplace of dust generated in materials handling.

- Seal enclosed transitions to prevent the escape of dust.
- Patch or turn worn spouts to extend their useful life.
- Seal couplings on spouts.
- Caulk split ring connectors and check them regularly to verify effective seals.
- Clean floors and dust collecting surfaces frequently. This simple action reduces the reintroduction of dust into the air, by air currents.
- Reduce the effective area of openings to contain dust inside hoppers and transitions.
- Fit receiving pits with self-closing, gravity-activated grates. These devices effectively limit the escape of dust released from the pit during receiving operations. Although expensive, they also reduce the air flow rate required to maintain adequate face velocity across the grate for dust control.

These are just some examples of measures that can reduce dust concentrations in grain-handling facilities. Not all are possible in existing facilities but all are extremely cost effective when measured against the cost of dust collection systems or the potential health costs of uncontrolled dust.

7.2 Dust collection systems

Excellent literature is available to the designer of dust collection systems. Refer to publications such as *Design of Industrial Exhaust Systems*, published by Alden and Kane Industrial Press; *Industrial Ventilation*, published by the American Conference of Governmental Industrial Hygienists; and the *ASHRAE Handbook*, published by the American Society of Heating, Refrigeration, and Air Conditioning Engineers.

When designing dust control systems, the concept of capture velocity is important. This is the air velocity required to prevent the escape of dust particles through an opening. For grain, the capture velocity is 1.0–1.5 m/s over the open area. Double these values if extreme cross drafts are present.

Table 5 presents a guide to dust control air flow rates for various components of equipment handling or processing materials. Air flow can vary considerably depending on the degree of enclosure, the flow rate of material, and the tendency of the grain to generate dust. For all dust collection systems, though, the minimum air velocity is 18 m/s.

8 SAMPLE PROBLEMS

Solving ventilation and low-pressure pneumatic conveying system problems involves many factors. First determine the operating constraints of the system, including:

- air volumes required
- conveying capacity required
- flow velocity
- physical layout and locations of components
- characteristics of material being conveyed
- atmospheric conditions on site

Table 5 Dust control air flow requirements

Component	Air flow
Bins	260 L/s per bin
Bucket elevators	510 L/s per m^2 cross-section
Floor dumps	1020 L/s per m^2 open face area
Mixers	
0.5 t	140 L/s
1.5 t	320 L/s
Over 1.5 t	450 L/s
Purifiers	150–200 L/s per m^2 of screen area
Scales	
200 L	135 L/s
350 L	190 L/s
400 L	290 L/s
Screw conveyors	95 L/s, ducts of 10 m outside circumference

Source: *Industrial ventilation, a manual of recommended practice.*

Use this information to determine the energy required to move the clean air and the conveyed material and, ultimately, to size the various components of the system.

For both clean air and conveyed material, consider these pressure loss factors:

- entry loss
- acceleration loss
- straight pipe loss
- loss at pipe bends
- collector loss
- total system pressure loss

8.1 Entry loss (H_e)

$$H_e = F \times P_v$$

where F = entry loss factor (see Appendix 1)

P_v = velocity pressure (Pa)

8.2 Acceleration loss (H_a)

For clean air

$$H_a = P_v$$

For conveyed material

$$H_a = 2.25 \times R_w \times P_v$$

where 2.25 = design factor, as set by the MikroPul Corporation

R_w = ratio of mass flow rate of solids to air

8.3 Straight pipe loss (H_p)

For clean air

$$H_p = L_e \times F_l$$

For conveyed material

$$H_p = F_m/F_a \times F_l \times L_e$$

where L_e = equivalent length of pipe (m)

F_l = friction loss (Pa/m)

F_m = friction loss of mixture

F_a = friction loss of air

In calculating the lengths of low-pressure pneumatic conveyors, add 15–20% to the length of vertical runs.

8.4 Loss at pipe bends (H_b)

For clean air

H_b = equivalent length of straight pipe (see Appendix 2)

For conveyed material

$$H_b = N \times [1+2(F_m/F_a)-1)] \times F_b \times P_v$$

where N = number of bends

F_b = fraction of velocity pressure (see Appendix 2)

8.5 Collector loss (H_c)

Collector loss varies depending on the collector. To estimate loss in most cases, use 2 to 4 times the velocity pressure. For fabric collectors or filters, however, refer to manufacturers' data.

For losses downstream of the collector, base calculations on clean air and include the effects of weather hoods, silencers, flow meters, and other similar components.

Treat systems with accessories used to condition the air, such as heaters, coolers, or dehumidifiers, as if they were clean air systems.

8.6 Total system pressure loss

Total system pressure loss is equal to the sum of all the individual losses, from the system inlet to the system discharge. This parameter applies to both the clean air and material.

8.7 Sample calculations for dust collection systems

Consider a simple system withdrawing dusty air from a floor dump pit measuring 1 m × 3 m (Fig. 24). Because dust collection systems handle essentially only clean air, ignore the mass of the airstream and the material it contains.

Use data from Table 5 to calculate the air flow rate to be extracted (Q).

$$Q = 1\text{ m} \times 3\text{ m} \times 1020\text{ (L/s)/m}^2$$

$$= 3060\text{ L/s}$$

Fig. 7 illustrates friction loss data for various duct diameters and velocities. For a duct velocity of 20 m/s and duct diameter of 500 mm, the friction loss is 8 Pa/m.

In this case

$$P_v = (v/1.29)^2$$

$$= 240\text{ Pa}$$

where P_v = velocity pressure (Pa)

From Appendix 1 the average coefficient of entry (C_e) at the pit for a tapered hood is 0.90.

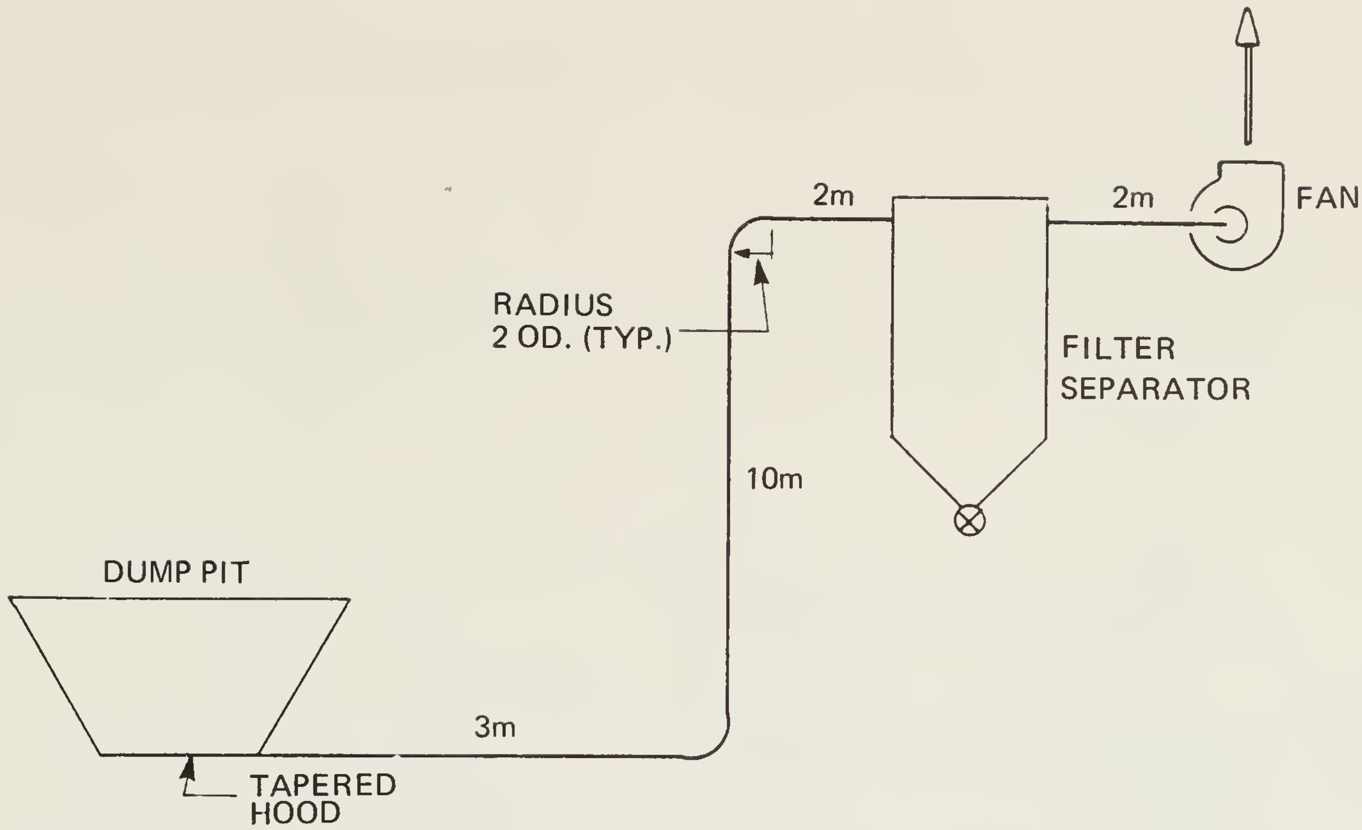


Fig. 24. Dust collection system.

To calculate entry loss (H_e)

$$H_e = \frac{1 - C_e^2 \times P_v}{C_e^2}$$

$$= \frac{1 - 0.90^2 \times 240}{0.90^2}$$

$$= 56 \text{ Pa}$$

From Appendix 2 the length of straight pipe equivalent to two elbows is

$$2 \times 9.2 = 18.4 \text{ m}$$

For a filter-type dust collector, pressure loss is 1500 Pa (see section 4.16 "Fabric collectors").

The equivalent length of pipe is

$$17 + 18.4 = 35.4 \text{ m}$$

Calculate pipe loss (H_p)

$$H_p = L_e \times F_1$$

$$= 35.4 \text{ m} \times 8 \text{ Pa/m}$$

$$= 283.2 \text{ Pa}$$

Calculate total system pressure.

$$\text{system pressure} = P_v + H_e + H_p + H_c$$

$$= 240 + 56 + 283.2 + 1500$$

$$= 2079.2 \text{ Pa}$$

Select a fan to deliver air at 3060 L/s at a pressure of 2079.2 Pa.

To power this system

$$\text{power} = \frac{\text{air flow rate} \times \text{total pressure}}{10^6 \times \text{fan efficiency}}$$

$$= \frac{3060 \text{ L/s} \times 2079.2 \text{ Pa}}{10^6 \times 0.65}$$

$$= 9.8 \text{ kW}$$

The mechanical efficiency for most centrifugal fan operating points is 0.50 to 0.65. Increasing the duct diameter between the collector and the fan could reduce power consumption. In this case, use a filter type of dust collector and a backward-curve or an airfoil centrifugal fan.

8.8 Calculations for a low-pressure pneumatic conveyor

Convey 5.0 t/h of wheat at standard atmospheric conditions (Fig. 25). The approximate velocity required to move the wheat in this example is 20–30 m/s (Table 3). Select 25 m/s.

For low-pressure pneumatic conveying, the maximum solids-to-air mass ratio is approximately 1.0. Because this factor is difficult to establish, often trial and error design methods are required. For example, initial calculations may call for a pressure out of the range of the low-pressure system; so the system parameters must be adjusted to handle a lower ratio of solids to air.

Aside from the pressure limitations, cost considerations of the system components direct the design.

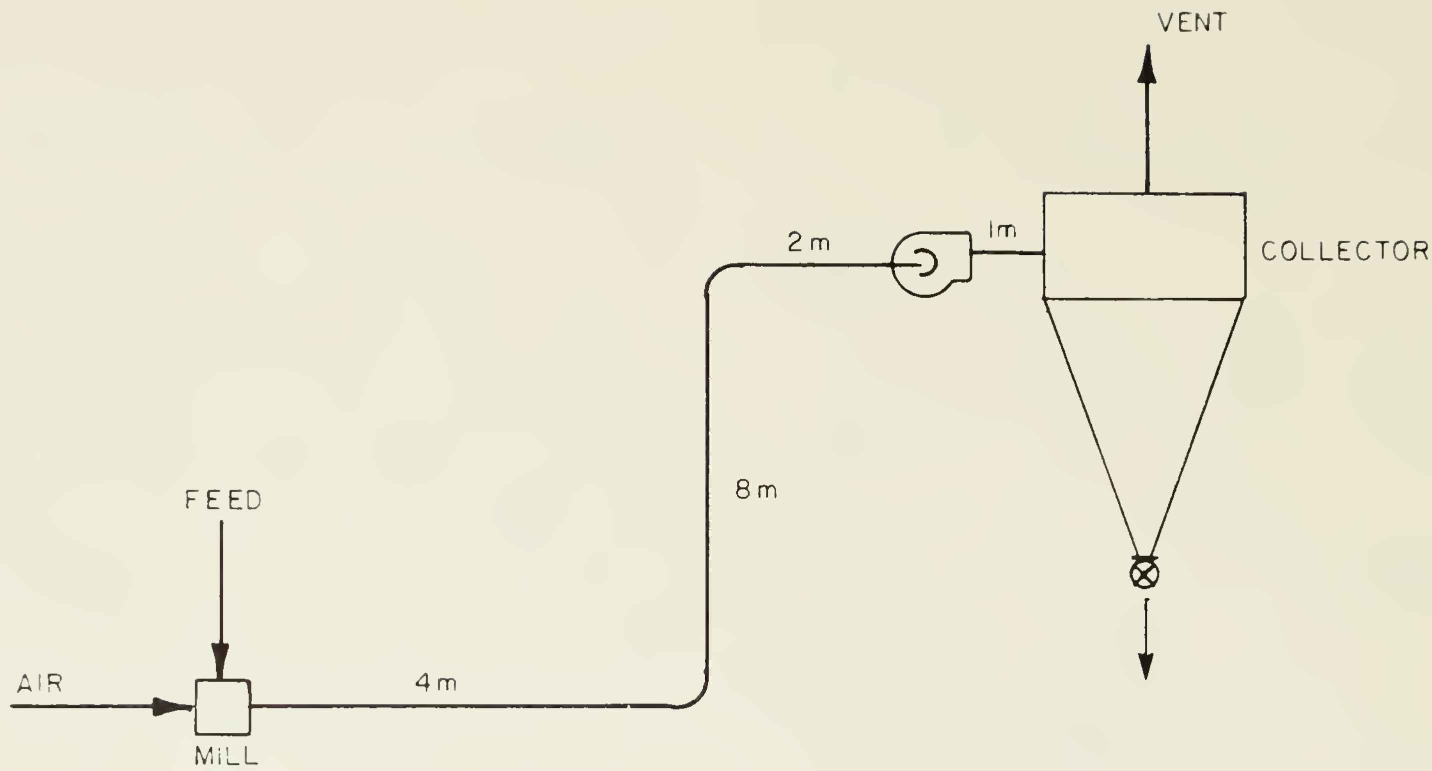


Fig. 25. Low-pressure pneumatic conveying system.

To determine air flow rate required

$R_w = M_m/M_a$

hence $M_a = M_m/R_w$

where M_m = mass flow rate of material (kg/s)

$$= \frac{5\ \text{t/h} \times 1000\ \text{kg/t}}{3600\ \text{s/h}}$$

$= 1.4$ kg/s

M_a = mass flow rate of air (kg/s)

$= 1.4$ kg/s

The air flow rate (Q) with standard air at 1.2 kg/m^3

$Q = 1.4/1.2$

$= 1.17\ m^3/s$

$= 1170$ L/s

From the chart of friction loss (Fig. 7), select a 225-mm duct and a velocity of 25 m/s with friction loss (F_l) equal to 30 Pa/m.

Under these conditions, the velocity pressure

$P_v = (v/1.29)^2$

$= (25/1.29)^2$

$= 376$ Pa

From Appendix 3, the ratio of the friction loss of the material (F_m) with the friction loss of air (F_a) for a solids-to-air mass ratio (R_w) of 1.0

$F_m/F_a = 1.35$

Calculate the system equivalent length.

Elbow equivalent length for radius

$= 2.5 \times$ diameter (Appendix 2)

2.9 m $\times$ 2 elbows $= 5.8$ m

Vertical equivalent length:
$1.15 \times 8\ \text{m} = 9.2$ m

Horizontal equivalent length
$= 7.0$ m

Total equivalent length (L_e)

L_e (clean air) $= (5.8 + 8.0 + 7.0)$

$= 20.8$ m

L_e (material) $= (9.2 + 7.0)$

$= 16.2$ m

Consider separately the pressure losses due to bends in the pipes for the material components.

Calculations for system material loss follow.

Acceleration pressure loss (H_a)

$H_a = 2.25 \times R_w \times P_v$

$= 2.25 \times 1.0 \times 376$

$= 846$ Pa

Straight pipe loss (H_p)

$H_p = (F_m/F_a) \times L_e \times F_l$

$= 1.35 \times 16.2\ \text{m} \times 30$ Pa/m

$= 656$ Pa

When estimating losses due to material friction, treat the loss at bends separate from straight pipe loss.

Bend loss (H_b)

$$H_b = N[1+2(F_m/F_a-1)] \times F_b \times P_v$$
$$= 2[1+2(1.35-1)] \times 0.22 \times 376$$
$$= 281 \text{ Pa}$$

System material pressure loss

$$H_m = H_a + H_p + H_b$$
$$= 846 + 656 + 281$$
$$= 1783 \text{ Pa}$$

Use the following formulas to calculate the clean air pressure loss for the system.

Entry loss (H_e) for a direct branch-booth type of hood, where $F = 0.50$

$$H_e = F \times P_v$$
$$= 0.50 \times 376$$
$$= 188 \text{ Pa}$$

Clean air acceleration loss (H_a)

$$H_a = P_v$$
$$= 376 \text{ Pa}$$

Straight pipe and elbows loss (H_p)

$$H_p = L_e \times F_l$$
$$= 20.8 \text{ m} \times 30 \text{ Pa/m}$$
$$= 624 \text{ Pa}$$

Collector loss (H_c)

$$H_c = 2P_v$$
$$= 2 \times 376$$
$$= 752 \text{ Pa}$$

Clean air loss (H_{ca})

$$H_{ca} = H_e + H_a + H_p + H_c$$
$$= 188 + 376 + 624 + 752$$
$$= 1940 \text{ Pa}$$

Total system pressure losses (H)

$$H = H_m + H_{ca}$$
$$= 1783 + 1940$$
$$= 3723 \text{ Pa}$$

Select a system to deliver 1170 L/s of air at 3723 Pa. The power required for this system

$$\text{power} = \frac{1170 \text{ L/s} \times 3723 \text{ Pa}}{10^6 \times 0.65}$$
$$= 6.7 \text{ kW}$$

8.9 Calculations for medium- and high-pressure pneumatic systems

The design of medium- and high-pressure systems is basically the same as for low-pressure systems, with some exceptions. For medium- and high-pressure systems, choose air velocities sufficient to convey the material without blocking the pipes. However, keep the velocity low enough to limit damage to the grain. Solids-to-weight ratios are higher for medium- and high-pressure systems, yet little data exist for the optimum ratios and the required velocities. In addition, pay attention to the effects of the large amounts of solids in the airstream.

Several situations contribute to air flow losses in medium- and high-pressure systems:

- acceleration of air to the carrying velocity
- friction due to air flow in the pipes
- material flow
- acceleration of the material
- support of solids in the vertical flow
- material friction in the pipes
- dynamic losses in bends

When calculating airstream losses consider, too, the compressibility of air.

For the most part, designing medium- and high-pressure systems requires a great deal of experience. However several publications can assist in this area (see "References").

REFERENCES

ASHRAE Handbook, S.I. Edition. Atlanta: American Society of Heating, Refrigeration, and Air Conditioning Engineers (ASHRAE), 1985.

Design of Industrial Exhaust Systems. New York: Alden and Kane Industrial Press, 1939.

Dominion Fire Commissioner. *Standard for grain elevators.* Ottawa: Department of Public Works, 1978.

Evaluation Report: John Deere 6500 Forage Blower. Report No. E0478B. Humboldt, Sask.: Prairie Agricultural Machinery Institute, 1979.

Evaluation Report: New Holland 28 Forage Blower. Report No. E0478A. Humboldt, Sask.: Prairie Agricultural Machinery Institute, 1979.

Fan Application Manual: Fans and Fan Systems. Arlington: Air Moving and Conditioning Association Inc., 1973.

Industrial Catalogue. Bethlehem: Fuller Company.

Industrial Catalogue. Muncy: Koppers Company, Inc., Sprout Waldron Division.

Industrial Catalogue: Air Compressors. Mississauga: Compair Canada Inc., 1984.

Industrial Ventilation, a Manual of Recommended Practice. Lansing: American Conference of Governmental Industrial Hygienists, 1982.

Kleis, R.W. 1955. *Operating characteristics of pneumatic grain conveyors.* Agric. Exp. Stat. Bull. 594.

Noyes, R.T.; Pfieffer, W.E. 1985. *Design procedures for pneumatic conveyors in agriculture.* ASAE paper no. 85-3507.

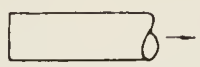
Reference data and tables. Bulletin RD-1. Summit: MikroPul Corporation, 1986.

Rider, A.R.; Barr, S.D. *Fundamentals of machine operation – hay and forage harvesting.* Moline: John Deere Service Publications, 1976.

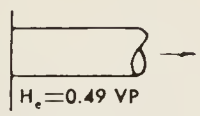
Stepanoff. *Gravity flow of bulk solids and transportation of solids in suspension.* New York: John Wiley and Sons.

Clausthal-Zellerfeld. 1982. *Bulk solids handling.* Int. J. Storing Handl. Bulk Mater. 2(3).

APPENDIX 1. HOOD ENTRY LOSS FORMULAS

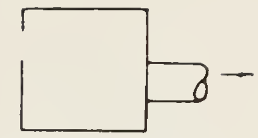


$H_e = 0.93$ VP
PLAIN DUCT END
$C_e = 0.72$

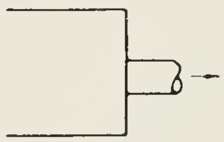


$H_e = 0.49$ VP
FLANGED DUCT END
$C_e = 0.82$

$H_e = 1.78$ VP
SHARP-EDGED ORIFICE
$C_e = 0.60$



$H_e = 2.3$ VP DUCT; OR BETTER: 1.78 VP ORIFICE + 0.49 VP Duct
ORIFICE PLUS FLANGED DUCT
(MANY SLOT TYPES)
$C_e = 0.55$



$H_e = 0.50$ VP
DIRECT BRANCH - BOOTH
$C_e = 0.82$

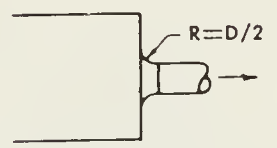
GENERAL FORMULAS

$$H_e = \frac{1 - C_e^2}{C_e^2} \times VP$$

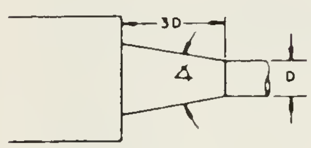
$$F = \frac{1 - C_e^2}{C_e^2}$$

$$H_e = F \times VP$$

F = ENTRY LOSS FACTOR
VP = VELOCITY PRESSURE IN DUCT
SP = STATIC PRESSURE AT THROAT,
H_e = ENTRY LOSS,
C_e = COEFFICIENT OF ENTRY

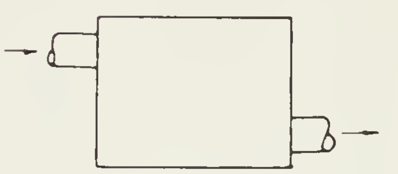


$H_e = 0.06$ VP TO 0.10 VP
BOOTH PLUS ROUNDED ENTRANCE
$C_e = 0.97$



BOOTH PLUS TAPERED ENTRANCE

	ENTRY LOSS		ENTRY COEFFICIENT	
	ROUND	RECTANGULAR	ROUND	RECTANGULAR
15°	0.15 VP	0.25 VP	0.93	0.89
30°	0 08 VP	0 16 VP	0.96	0.93
45°	0.06 VP	0.15 VP	0.97	0.93
60°	0 08 VP	0.17 VP	0.96	0.92
90°	0.15 VP	0.25 VP	0.93	0 89
120°	0.26 VP	0.35 VP	0.89	0 86
150°	0 40 VP	0.40 VP	0.84	0.82



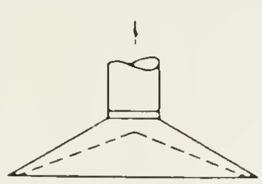
$H_e = 1.5$ VP
TRAP OR SETTLING CHAMBER
$C_e = 0.63$ (approx.)

ENTRY LOSS FOR COMPLICATED HOOD SHAPES:

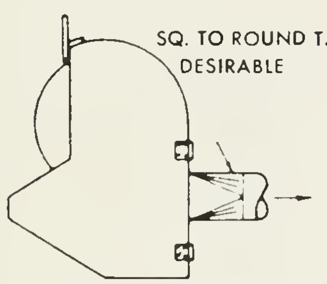
1. BREAK HOOD INTO SIMPLE COMPONENTS
2. CALCULATE H_e FOR EACH COMPONENT
3. ADD VALUES OF H_e.

MISCELLANEOUS VALUES

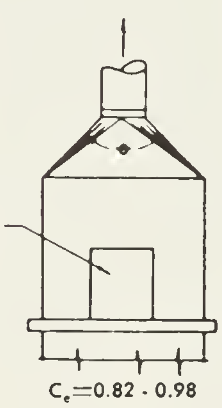
HOOD	ENTRY LOSS, F
ABRASIVE BLAST CHAMBER	1.0
ABRASIVE BLAST ELEVATOR	2.3
ABRASIVE SEPARATOR	2.3
ELEVATORS (ENCLOSURES)	0.69
FLANGED PIPE PLUS CLOSE ELBOW	0.8
PLAIN PIPE PLUS CLOSE ELBOW	1.60
TUMBLING MILLS (VARIES WITH DESIGN OF MILL)	AV. 2.0



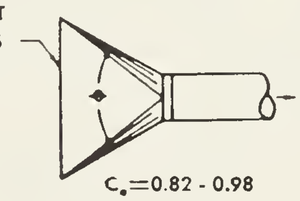
$H_e = 1.0$ VP
DOUBLE (INNER CONE) HOOD
$C_e = 0.70$ (approx.)



$H_e = 0.65$ VP
STANDARD GRINDER HOOD
$C_e = 0.78$



$C_e = 0.82 - 0.98$



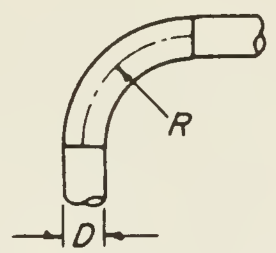
$C_e = 0.82 - 0.98$

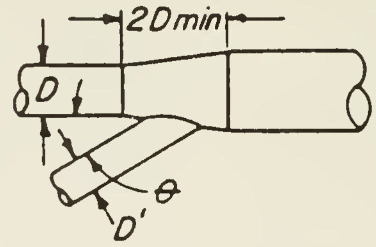
TAPERED HOODS

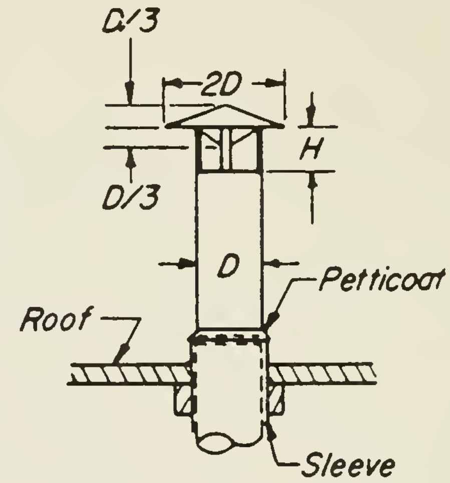
FLANGED OR UNFLANGED; ROUND, SQUARE OR RECTANGULAR.
◆ IS THE MAJOR ANGLE ON RECTANGULAR HOODS.

Source: *Reference data and tables. Bulletin RD-1.*

APPENDIX 2. EQUIVALENT RESISTANCE METRES OF STRAIGHT PIPE



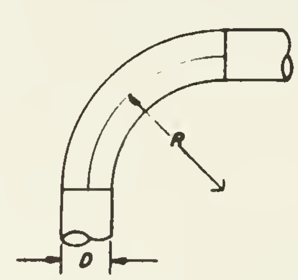




Not recommended

Pipe in mm	90° Elbow * Centerline Radius			Angle of Entry		H, No of Diameters		
	1.5D	2.0D	2.5D	30°	45°	1.0H	0.75H	0.5H
75	1.4	0.9	0.7	0.5	0.9	0.3	0.5	2.0
100	2.0	1.3	1.1	0.8	1.3	0.5	0.8	3.4
125	2.6	1.7	1.4	1.1	1.7	0.6	1.1	4.4
150	3.2	2.2	1.8	1.4	2.2	0.8	1.4	5.5
175	3.9	2.6	2.2	1.7	2.6	0.9	1.7	6.6
200	4.6	3.1	2.5	2.0	3.1	1.1	2.0	7.8
250	6.0	4.0	3.3	2.6	4.0	1.4	2.6	10
300	7.4	5.0	4.1	3.2	5.0	1.8	3.2	13
350	8.9	6.0	5.0	3.8	6.0	2.1	3.8	15
400	10	7.0	5.8	4.5	7.0	2.5	4.5	18
450	12	8.1	6.7	5.2	8.1	2.8	5.2	21
500	14	9.2	7.6	5.9	9.2	3.2	5.9	23
600	17	11	9.5	7.3	11	4.0	7.3	29
700	21	14	11	8.8	14	4.8	8.8	35
800	24	16	13	10	16	5.7	10	41
900	28	19	15					
1000	32	21	18					
1200	39	26	22					
1400	47	32	26					
1600	55	37	31					
1800	64	43	36					
2000	72	49	40					

*For 60° elbows —— x.67
For 45° elbows —— x.5

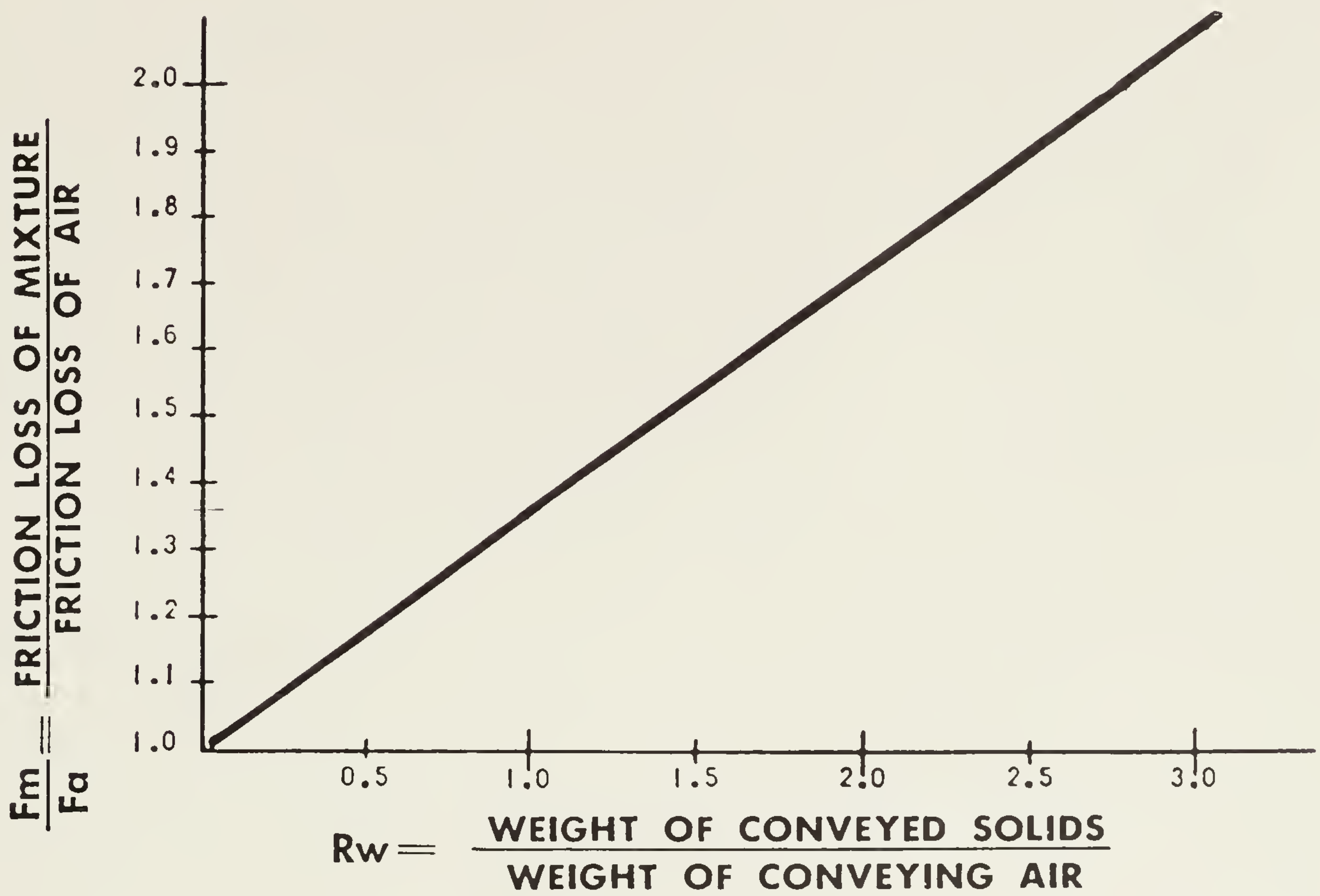


R, No. of Diameters	Loss Fraction of VP
2.75 D	0.26
2.50 D	0.22
2.25 D	0.26
2.00 D	0.27
1.75 D	0.32
1.50 D	0.39
1.25 D	0.55

ROUND ELBOWS

Source: *Industrial ventilation, a manual of recommended practice.*

APPENDIX 3. RATIO OF FRICTION OF MIXTURE TO FRICTION OF AIR ALONE



Source: *Reference data and tables. Bulletin RD-1.*